9 Ship design, construction and operation

Contents

The various Sections of this Chapter have been taken from the following books and sources, with the permission of the authors:

Eyres, D.J. (2007) *Ship Construction*. 6th Edition. Butterworth-Heinemann, Oxford, UK. [Sections 9.3.2, 9.3.3, 9.3.5, 9.4]

Molland, A.F. (2005) *Ship Design and Economics*. Lecture Notes. School of Engineering Sciences, University of Southampton, UK. [Sections 9.1, 9.2]

Schneekluth, H. and Bertram, V. (1998) *Ship Design for Efficiency and Economy*. 2nd Edition. Butterworth-Heinemann, Oxford, UK. [Section 9.6]

Shenoi, R.A. and Dodkins, A.R. (2000) Design of Ships and Marine Structures made from FRP Composite Materials, in Kelly. A. and Zweben, C. (eds), *Comprehensive Composite Materials*, Vol. 6, Elsevier Science Ltd, Oxford, UK. [Section 9.3.4]

Watson, D.G.M. (1998) *Practical Ship Design. Elsevier Science*, Oxford, UK. [Section 9.5]

9.1 Introduction

This Chapter provides a broad overview of ship design, construction and operation. In the first sections, the basic practical aspects of deriving the technical ship dimensions, masses, stability and body plan are described. The next sections describe the materials used in ship construction, the effects of corrosion, and a brief description of the ship construction process. The economics of ship operation are described, indicating the interactions with technical design. Finally, optimization applied to ship design and operation is described and how optimization may be used to achieve the most suitable design.

9.2 Ship design

9.2.1 Overview

9.2.1.1 *General*

A ship is a complex vehicle. Its production requires the involvement of a wide range of engineering disciplines. Ship design is not an exact science but embraces a mixture of theoretical analysis and empirical data accumulated from previous successful designs. Due to the complex interrelationships between features of the technical design, and the construction of the ship and its operation, the final ship design will often represent a compromise between conflicting ship requirements.

The development of the overall ship design and its production cannot normally be treated in technical isolation as operational requirements have to be considered. For example, the ship will often form part of a through transport system; this may range from sophisticated container systems with dedicated ships operating between specified ports, or ferries and RO/RO vessels relying on a regular wheeled through cargo, to tramp vessels on non-regular schedules which rely on carrying various types of cargo between various ports. A review of some of these ship types is given in Chapter 2.

The route and its environment, type of cargo, quantity to be moved, value of the cargo and port facilities are typical features which will be considered when evolving the size, speed and specification of a suitable ship (or ships). Specific service requirements will be similarly considered when evolving vessels such as warships, passenger ships or fishing vessels.

Shipowners operate ships to make a satisfactory profit on their investment. The evolution of a technical design can therefore be considered as a component part of an overall economic model. In evolving a ship design it is therefore necessary to assess the operating requirements and the environment in which the vessel is to operate, to evolve the feasible technical design and to economically justify the viability of the proposal.

In an overall final design process the design objectives have to be clearly identified and constraints in the process incorporated. The following discusses some of the alternative objectives:

Design for functionability, or capability: this is a pre-requisite without which the ship does not fulfil its role, whether it be a warship or a large tanker.

Design for efficiency and economy: this is normally also a pre-requisite and might take several forms including designing to minimize running costs, maintenance costs, cranage/turnround time for container ships, or turnround time for ferries (e.g. manoeuvring), all with a view to improving the overall efficiency of the operation.

Design for production: In this case producibility is important, and savings in construction costs may be assessed, Kuo *et al.* (1984), Andrews *et al.*(2005). In this case, the analysis may, for example, be trading increases in steel mass (and hence decrease in deadweight) against decreases in production costs.

Design for maintenance: this will often amount to increase in space and improved access for maintenance of tanks or machines. This might entail accepting surplus volume and an increase in ship first cost.

Design for the environment: aspects may include pollution, emissions, noise and wave wash. These objectives are becoming increasingly important. Some of these aspects are covered in MARPOL.

Design for disposal, or scrap: this is becoming more important in the design process, whereby ease of disposal (e.g. cutting up hull, or removing machinery) is taken into account.

Each objective is important in its own right. Whilst achievement of all the objectives is desirable, but unlikely, some weighting as to the relative importance of the various objectives will normally be necessary.

The following Sections consider the practical aspects of evolving the technical design model, bearing in mind operational patterns and requirements, and its extension in Section 9.5 to include economic considerations and evaluations. Assessments of alternative design methodologies and philosophies are not carried out; these can be studied further in texts and references such as Gillmer (1977), Watson (1998), Schneekluth and Bertram (1998), Eames and Drummond (1977) and Andrews (1981, 1998, 2007).

9.2.1.2 *Ship design process*

The ship design process may be broken down broadly into two stages:

(i) Conceptual and/or preliminary design
(ii) Detailed or tender or contract design

The principal ship dimensions and power to meet the intended service will be evolved at stage (i). If the results of stage (i) are technically and economically viable then stage (ii) will follow. The development of the detailed requirements up to contract stage should not normally have a significant effect on the basic particulars evolved at stage (i).

This section is concerned, in the main, with the evolution of the preliminary ship design and its evaluation. Involvement in detailed and constructional aspects is limited to a brief overview in Section 9.4.

The preliminary design process will normally take the form of a techno-economic appraisal, using a fundamental engineering economy approach.

The increase in effort to improve efficiency has led to an increasing use of economic investigation. Whilst a primary and traditional function of the ship designer or naval architect is to derive a feasible technical design, it is unlikely that this will be achieved in technical isolation without taking account of economic considerations, either directly or indirectly.

The application of engineering economics to ship design is basically the conversion of the marine transport requirements into a range of feasible ship designs which must then be evaluated for their technical and economic performance.

The overall flow path shown in Figure 9.1 can thus be established.

Many of the techno-economic evaluations amount to an investigation of the trade-off between first and operating costs; it is important to note that the 'best' design need not necessarily be of lowest first cost, but that which shows the most profitable combination of first and operating costs over the life cycle.

9.2.2 Technical ship design

9.2.2.1 *Principal requirements*

The principal requirements of a technical ship design may be summarized as follows:

1. Is adequate in size and arrangement for intended service	implies ability to carry a specified volume of cargo and have adequate space for machinery, fuel and crew etc.
2. Floats at correct draught	implies sum of weights of lightship and deadweight equals force due to buoyancy (function of ship form)
3. Floats upright	implies adequate stability
4. Achieves correct speed	implies satisfactory estimates of resistance and propulsive power (plus margins) and installation of suitable engine(s).
5. Is structurally safe/ sound	implies structural design with the ability to withstand forces in the marine environment; typically built to the requirements of a classification society
6. Meets requirements for manoeuvring, coursekeeping and seakeeping	implies choice of suitable hull form
7. Meets international standards of safety and reliability	meets requirements of IMO

The derivation of a feasible technical design will take the form of an 'iterative process of analysis and synthesis'; i.e. is a repetitive process whereby the design is resolved into simple elements and relevant calculations made, after which the elements are combined into the total ship design.

For example, for a deadweight determined design, items 1 to 4 might be modelled as in Figure 9.2.

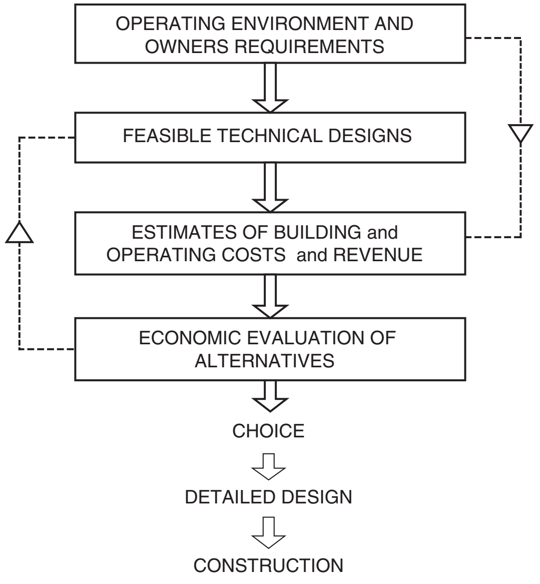


Figure 9.1 Overall flow path.

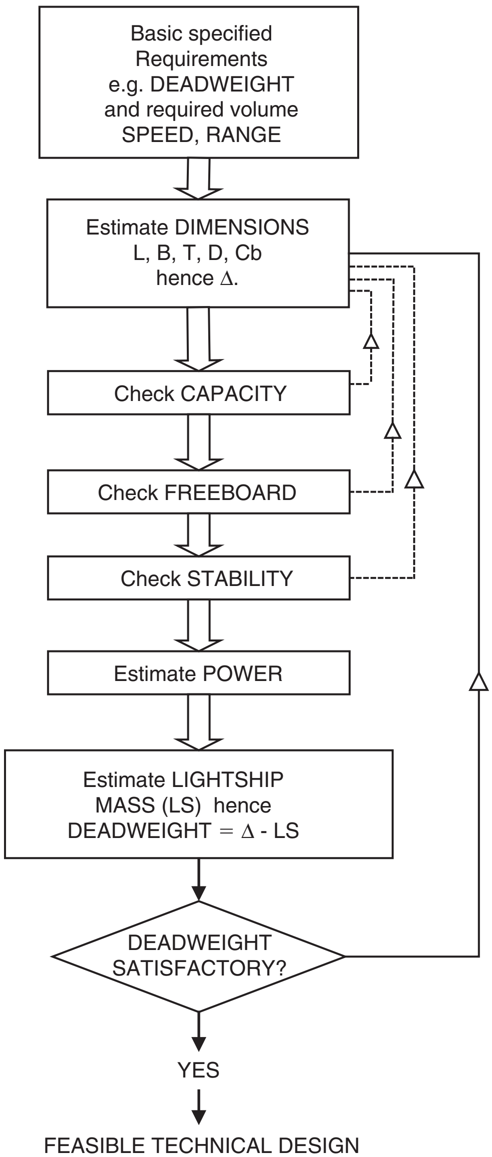


Figure 9.2 Preliminary design path.

9.2.2.2 *Specification*

The owner's operational requirements need to be established, which then allows the development of a basic specification.

An owner might typically be seeking a new ship design to suit one or a combination of the following alternatives:

- Replacement or conversion of old vessels
- Expansion or modification of services on an existing route in an effort to enlarge the participation
- Development of a new service or carrying a different kind of cargo on an existing route aimed at capturing an increased percentage of the trade
- Development of a new service on a new route

In each situation, the owner is faced with decisions concerning the number of ships required, their type, size and speed.

Before the design process can be initiated, the basic technical data relating to the operational requirements have to be defined and specified, or derived or assumed if several alternatives are to be investigated.

For a deadweight carrier, the basic specification requirements would typically be as follows:

Deadweight:	Will become a variable if alternative ship sizes are to be investigated.
Speed:	Possible 'hydrodynamic optimum' speed for particular ship length, or may be dependent on nature/value of cargo, e.g. perishable fruit or passengers. Will be specified, or alternatives may be investigated.
Capacity:	Specified to suit the design cargo stowage rate (m^3/tonne).
Range:	Range and route, length of voyage – fuel capacity, route – weather conditions/ power margins.
Stability:	Minimum requirements usually specified for most onerous conditions (e.g. loaded arrival is not uncommon).
Strength:	Minimum requirements usually specified to be those of one of the classification societies.
Manoeuvrability:	Specification may relate to rudders, etc. lateral thrusters, tug dues, etc. mooring arrangements.
Trim:	Normally required by stern.
Dimensional Constraints	e.g. specific breadth or width or draught limitations, such as canal or port limits in breadth and/or draught.
General Arrangement:	To meet the needs of specified crew, cargo handling, passenger accommodation.
Statutory Regulations:	To meet requirements.

At the *preliminary* design stage, for a deadweight carrier, it is often suitable to treat the following as primary requirements:

- DEADWEIGHT
- SPEED
- RANGE

And to treat the following as 'checking' or constraint requirements:

CAPACITY
STABILITY
FREEBOARD
Plus others if necessary

It can be noted that this procedure covers a large proportion of merchant ship types, but an alternative known as a *capacity approach* is necessary in the case of *capacity carriers* such as passenger ships, ferries, warships and container ships. In such cases the pre-requisite is to contain a certain capacity or volume rather than to lift a particular deadweight. The *CAPACITY* or *SPACE DESIGN* approach is discussed later in Section 9.2.4.

9.2.3 Deadweight determined designs

A *deadweight design approach* is based on equating the sum of the component masses of the vessel to its displacement. It is applicable to the majority of ship types including tankers, bulk and ore carriers, and most cargo vessels.

9.2.3.1 *Deadweight and dimensions*

(a) Deadweight (DW):
Includes cargo, fuel, FW, stores crew and effects. Cargo is the only component of deadweight, which will earn revenue, hence other items of deadweight should be kept to a minimum.

(b) Lightship mass (LS):
Condition is that of a ship when ready to put to sea, but without cargo, fuel, stores and provisions. The primary components of lightship mass are steel, outfit and machinery.

(c) Displacement (Δ):
Total ship mass: equals mass of water displaced, equals 1.025 L.B.T.C_B, and Deadweight = Displacement − Lightship

A primary aim is to design a ship with minimum Δ to meet the requirements of the owner, hence obtaining the most economical ship in respect of the machinery, fuel consumption and initial cost.

(d) Deadweight coefficient (C_D):

is defined as C_D = Total Deadweight/Displacement

and can be treated as a very approximate criterion or measure of 'efficiency' of the vessel.

A preliminary value of displacement can be determined from C_D, when the DW has been defined.

Typical values of C_D are as follows:

Cargo ships	0.65–0.75
Large tankers/Bulk	0.79–0.85
Ore	0.82
*Container	0.60
*Refrigerated cargo	0.55–0.60
*Passenger	0.35

*For these the predominant factor is that of space, hence C_D of little significance.

Note that C_D will vary with cargo type since bulky cargoes require greater volume (hence steel), hence C_D will be lower. Similarly a higher speed (for same DW) will involve increases in machinery mass hence in LS and reduction in C_D. Hence special care is needed in the use of this coefficient.

Derivation of dimensions:
(1) Length (L):
Usually a minimum consistent with speed and form; length is generally the most expensive dimension. A preliminary estimate of length may be made using:

$L = f(\nabla^{1/3})$, where the function depends on ship type.

$L/\nabla^{1/3}$ lies typically in the range 5.5 to 6.5 for cargo vessels and tankers and 6.5 to 8.5 for higher speed vessels and passenger ships.

(2) Breadth (B):
Has direct influence on stability. L/B has influence on hull resistance, hence power. L/B tends to be larger for faster ships.

L/B ratios for cargo ships lie within range 6–7 and for passenger ships with range 6.5–7.5.

Typical empirical values (from Watson and Gilfillian (1977)) are as follows:

$L/B = 4$	($L < 30$m)
$L/B = 4 + 0.025\,(L-30)$...	($L = 30$–130m)
$L/B = 6.5$	($L > 130$m)

(3) Draught (T):
B/T ratio related to hydrodynamic performance and stability

B/T for cargo ships typically varies between 2 and 2.5
B/T for pass. ships typically varies between 3 and 5
T related to D in respect of freeboard and T/D is typically 0.7–0.8 for cargo vessels, bulk carriers and tankers

(4) Depth (D):
Can be approximated from L/D ratio (related to strength of ship), or B/D ratio (related to stability)
L/D for cargo, tankers bulk carriers typically 12–13
B/D≅ 1.9 for DW carriers such as tankers and bulk carriers
B/D≅ 1.7 for stability limited capacity carriers

(5) Block coefficient (C_B):
For economical propulsion from a hydrodynamic point of view, length and fullness at a given speed are closely related. Typical approximate formulae relate C_B with $V/\sqrt{L}$ as follows.

$$C_B = a - b\, V/\sqrt{L} \qquad \text{[V knots, L metres]}$$

Where typical values of a and b are 1.23 and 0.395.

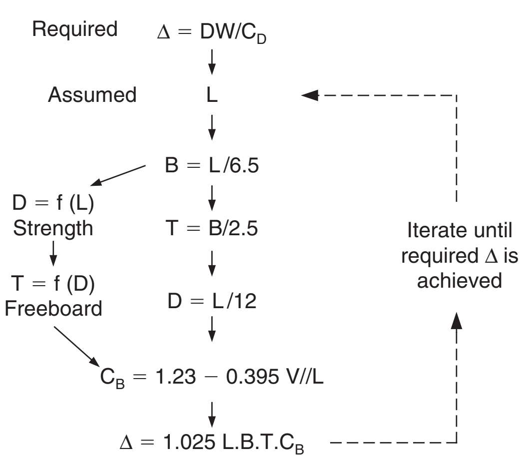


Figure 9.3 Flow chart for dimensions (deadweight approach).

(6) Derived dimensions:
A simple model to derive the principal dimensions for a given deadweight (DW) and speed (V), using functional relationships between the dimensions might be developed as shown in Figure 9.3.

L may be assumed, or approximated using $L = f(\nabla^{1/3})$ in first cycle.

9.2.3.2 *Cargo capacity check*

A deadweight design approach ensures that the correct *mass* of cargo can be carried. The volumetric *capacity* of the vessel must be such that the volume of the required mass can be contained.
Typical cargo stowage rates are as follows:

General cargo	1.4–1.7 m^3/tonne
Refrigerated cargo	1.8–2.0 m^3/tonne
Crude Oil	1.05 m^3/tonne (approx.)

In the design process, the required volumetric cargo capacity V_C can be estimated from a knowledge of the cargo deadweight and the stowage rate for the cargo type.

Definitions:
Moulded volume
Total internal hull volume to inside of shell plating.

Grain capacity
Taken to top of beams, inside shell plating and tank top and is moulded volume less actual volume of all obstructions such as structure. Grain capacity is typically about 1.5–2% less than moulded volume in the holds and about 3% less in double bottoms.

Bale capacity
Capacity taken to underside of beams, inside of frames, inside of beam knees etc. Bale capacity typically about 10–12% less than the grain capacity.

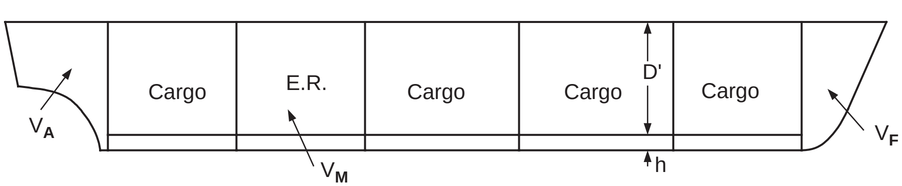


Figure 9.4 Capacity check.

Capacity check

At the detailed design stage the capacity check presents no problems as it can readily be estimated from the ship hull form and general arrangement.

At the preliminary stage approximate relationships have to be applied.

If the capacity of a suitable basis vessel is available, then a preliminary estimate for the new design can be made by scaling the dimensions. The method is more accurate if all the underdeck volume is used, i.e. including other non-cargo spaces such as machinery. Hence if G_1 is the total underdeck grain capacity for the basis vessel then the corrected total underdeck volume for the new design will be:

$$G_2 = G_1 \times L_2/L_1 \times B_2/B_1 \times D_2/D_1 \times C_{B2} \ @\ 0.85D/C_{B1} \ @\ 0.85D$$

(Depth used should take account of shear and double bottom)

The requirements of non-cargo spaces, such as machinery and accommodation etc. underdeck, for the new design proposal are then subtracted from G_2 to give the estimated cargo capacity.

If a suitable particular basis vessel is not available at the preliminary stage, a preliminary check may be made using a data base from similar ships. This may be carried out as follows, Figure 9.4:

Total Underdeck Volume $V_T = LBD'C_B'$

where D' allows for double bottom, hence $D' = D\text{-}h$ and $h = f(D)$; $C_B' =$ block coefficient at 80% D and is a criterion of fullness up to main deck with C_B' derived from C_B @ design draught T.

From the total volume V_T will be deducted:

(i) the volume of the machinery space V_M:
An approximate assumption is that the volume required by the main engine and auxillary machinery is a function of power, hence $V_M = f(\text{Power}) = f(\Delta^{2/3}\ V^3)$, where the function will depend on ship type, size and position of the machinery space.

(ii) the non-cargo volumes within the length but forward and aft of the cargo space $(V_F + V_A)$; these will be typically expressed as a percentage of the total volume V_T for a particular ship type

i.e. $(V_F + V_A) = f(LBD'C_B')$

Thus the cargo capacity may be expressed approximately in terms of the variables already determined at that stage in the design path i.e.

$$V_C = V_T - V_M - (V_F + V_A) = f_1(LBD'C_B') - f_2(\Delta^{2/3}V^3) - f_3(LBD'C_B')$$

where f_1, f_2 and f_3 would be obtained from similar basis ships.

9.2.3.3 *Summary of overall model: Deadweight approach*

The derivation of the dimensions can now be incorporated into an overall technical design model, as shown in Figure 9.5.

The model illustrated is simple, but lends itself to systematic variations in components of the design, e.g. methodical variation of deadweight, speed, dimensions, etc. The model shows functional relationships between the principal dimensions. It should be noted that all dimensions could be ‘free floating’ in the design procedure provided adequate constraints confine particulars to physical limitations, the power estimate is adequate to predict changes due to distorted dimensional relationships and data requirements are within the range of any empirical relationships used.

It should be noted that at this stage in the design process a design has been derived which is *feasible*, although it may not be the best for its intended purpose. It will be seen later that the model providing the derivation of the alternative feasible technical designs can be incorporated in a larger model in which economic evaluations of the alternatives can be carried out see Section 9.6.

9.2.4 Capacity (or space) determined designs

A *capacity design approach* is used where the dimensions are required to be determined (primarily) by the need to provide a requisite space. Examples include passenger ships, most naval vessels, cargo ships with high stowage rates (such as for meat, bananas and cars) passenger/car ferries and container ships.

9.2.4.1 *Cargo ships*

The capacity design approach for this vessel type is based on the equality: $V_C = V_H - V_S$, where $V_C =$ vol. available for cargo, $V_H =$ total underdeck volume and $V_S =$ volume for machinery and other essentials. V_C is known and V_S may be estimated from basis vessels and/or known power requirements. Hence dimensions based on V_H may be derived as shown in Figure 9.6.

9.2.4.2 *Passenger ships*

In this case it is necessary to consider the volume of the whole ship including erections. The main

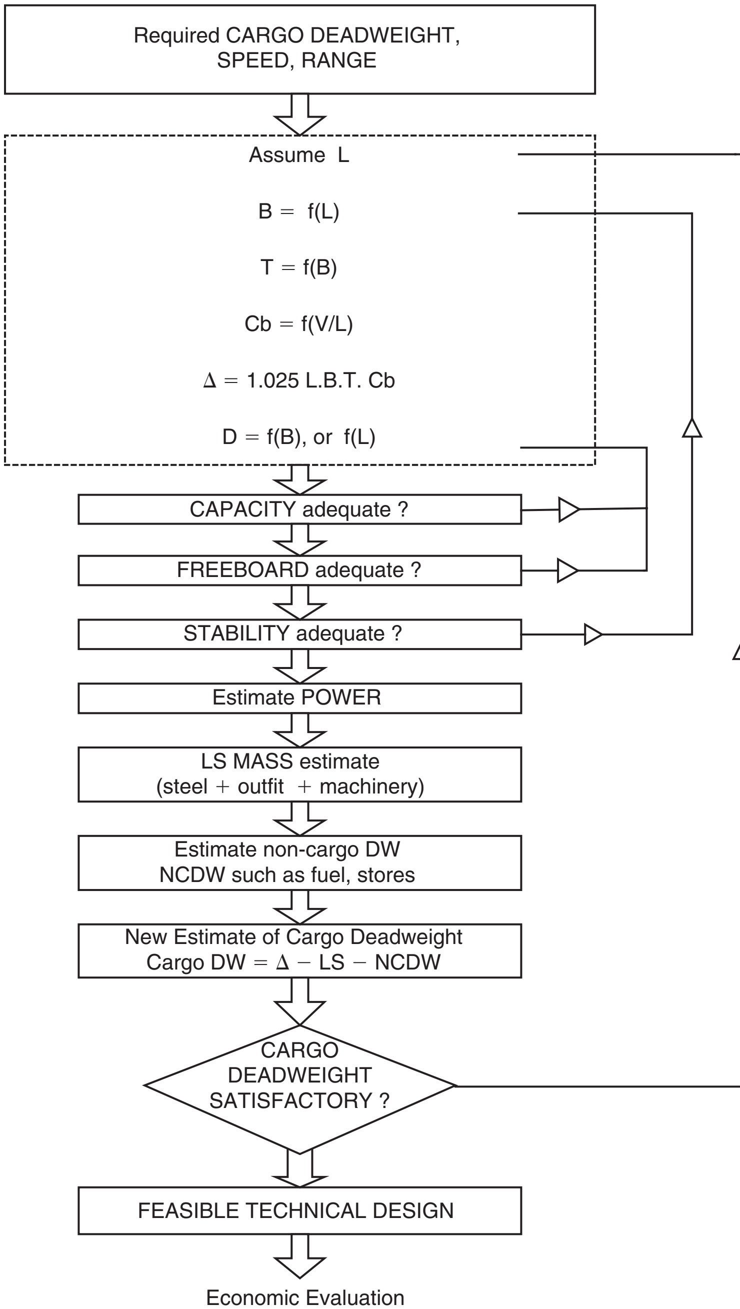


Figure 9.5 Preliminary design path.

problem is essentially one of calculating the volume required to permit the arrangement of the required passenger and crew spaces, machinery spaces etc., rather than the process of obtaining the actual dimensions to give the required volume, as described below.

The total volume requirement (V_T) is derived from a summation of the volumes of passenger accommodation (α pass. no.), public areas, crew accommodation (α crew no.), machinery (α power requirements) and volumes for fuel, FW, stores, etc. Values may be analysed from past ship data or

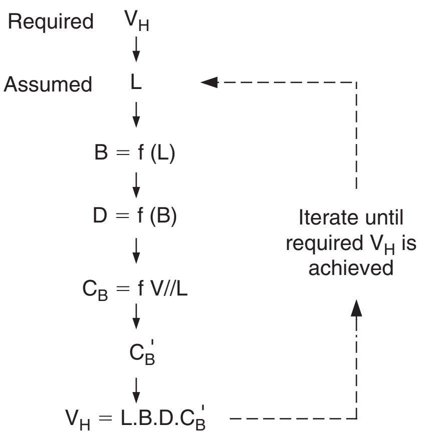


Figure 9.6 Flow chart for dimensions (capacity approach).

from sources such as Watson (1962) and Watson and Gilfillian (1977).

$$\text{Total volume } (V_T) = \text{Hull volume } (V_H) + \text{superstructure volume}$$

The superstructure is typically 30%–40% of total volume for passenger ships and current practice for a particular vessel type can be readily investigated.

Hence with total volume known, volume of hull (V_H) can be deduced.

In most cases, main hull volume V_H can be estimated on the basis:

$$V_H = L.B.D.C_B', \text{ where } C_B' = C_B \text{ @ } 0.8D$$
$$\text{e.g. } C_B' = C_B + (1 - C_B)(0.8D - T)/3T$$

Hence the required volume may be modelled as shown in Figure 9.6.

At this stage, the main hull profile and erections can be drawn to give the required volume and general arrangement. A *STABILITY* check (see Section 9.2.5) is necessary before any further detailed arrangements are developed. This is important in the case of the passenger ship since late changes (say in beam) may have a significant influence on the internal arrangement/layout of cabins, etc. – or on car lane widths in the case of large car ferries. Similarly, *FREEBOARD* (see Section 9.2.8) would be checked at an early stage in the case of large car ferries in order to site the freeboard deck and deck heights etc.

9.2.4.3 *Container ships*

These may be defined as 'linear dimensions' ships (e.g. see Watson (1962) and Watson and Gilfillian (1977)), further examples being St Laurence Seaway/Panama with breadth restrictions and some car ferries with B as a function of car lanes, etc.

Classical references on container ship design are Henry and Karsch (1966), Meek (1970) and Meek *et al.* (1972) and these provide good accounts of the basic design procedures and problems associated with the design of container ships. See also Section 9.6.6.2.

Container ships may be classified as capacity or space determined designs and their size is generally defined by their container capacity – e.g. 1500, 3000, 6000 or 10000 TEU. (Container sizes 20′ × 8′ × 8.5′ height or 40′ × 8′ × 8.5′). TEU = Twenty Foot Equivalent Unit (e.g. 40′ container = 2 TEU's).

Stowage rates for containers are typically max 20 tonnes/20′ container, but actual stowage rate is about 12 tonnes/container. For example, if the cargo deadweight is specified, then number of containers (n) can be derived.

Due to the high stowage rate of cargo in containers and containers in holds, this leads to the requirement for a large quantity of containers on deck. A decision is first made on number of containers on deck, e.g. say 6 deep in holds, 3 high on deck – hence approximately 2/3 of total n containers to be stowed below deck.

Hence ship hull dimensions may be designed around a capacity to contain say 2/3 n containers + machinery volume + double bottoms + peaks, etc.

Containers are best stowed in a rectangular space (say about midships), consequently machinery/accommodation is ideally placed, although not always feasible or desirable, see Meek (1970), and current designs tend to have machinery/accommodation about ¾ aft.

Beam:
B = f [container breadth + clearances + sufficient deck width outside line of hatches for required longitudinal and torsional strength]

Clearance between containers 9″–12″ for preliminary design.

Minimum strength width each side of hatches may be assumed to be about 10% B, i.e. about 20% overall.

Length:
Can be adjusted to give suitable dimensional ratios based on B. Length enclosing containers = f[containers length + clearances + bulkheads/stiffs].

In detail, Meek (1970) quotes length (or breadth) as being = f [container length + tolerance + structure to support cell guides + cell guide clearance + cell guide tolerance + adequate ship structure, etc].

Depth:
D = f [container depth + double bottom depth].

$D \alpha B$ is very important in view of stability requirements with deck cargo, i.e. static stability and dynamical re-wind effects – possible use of water ballast.

Note also development of hatchless container ships (cell guides running up above deck level) facilitating faster turn around time. (Removal/replacement of container ship pontoon hatch covers time consuming.)

9.2.4.4 *High speed passenger/vehicle ferries*

The derivation of the dimensions for these vessel types is usually based on areas (for given tween deck height) for given number of passengers and/or vehicles.

A description of the concept design and derivation of dimensions for these vessel types (currently for monohulls and catamarans) is given in Karayannis *et al.*(1999) and Molland *et al.*(2003), together with regression equations relating areas to passengers and/or vehicle numbers and $(L \times B)$ to required areas. Figures 9.7 and 9.8 and Table 9.1 from Molland *et al.* (2003), describe typical design flow paths and regression formulae for high speed ferries.

In this approach, described by Molland *et al.* (2003), the initial derivation of the dimensions is based on suitable values of the L/B ratio and $L \times B$ product, and hence a solution for L and B.

Hydrostatics/Hydrodynamics:
L/B is based on hull hydrostatic and hydrodynamic requirements and suitable assumptions for $L/\nabla^{1/3}$, C_B and B/T:

$$\frac{L}{B} = \left[\left(\frac{L}{\nabla^{1/3}}\right)^3 \cdot C_B \cdot \frac{T}{B}\right]^{1/2} \tag{9.1}$$

Areas:
The product $L \times B$ is based on required passenger and vehicle areas:

$$L \times B = f(A_p,\ A_V), \tag{9.2}$$

where $A_S = f(N_P)$, $A_P = f(A_S)$ and $A_V = f(N_V)$.

Suitable forms of these relationships, as well as ranges of the design parameters, are given in Table 9.1. The $L \times B$ product is derived using a three-step procedure as shown in Figures 9.7 and 9.8. This offers more flexibility in selecting the desired level of seating comfort and overall accommodation quality, which is achieved by appropriate adjustment of the passenger area relationships.

The solution for L becomes:

$$L = \left[(L \times B) \times \left(\frac{L}{B}\right)\right]^{1/2} \tag{9.3}$$

and B can then be derived from L/B, T from B/T and $\Delta = \rho.L.B.T.C_B$

For catamarans, L/B is derived as:

$$\frac{L}{B} = 1/\left[\frac{S}{L} + \frac{b}{L}\right] \tag{9.4}$$

where b is the breadth of a demihull and S the separation of the demihull centrelines.

In the case of catamarans, L/b is derived as:

$$\frac{L}{b} = \left[\left(\frac{L}{\nabla^{1/3}}\right)^3 \times C_B \times \frac{T}{b}\right]^{1/2} \tag{9.5}$$

in this case, B in Equation (9.1) is replaced by b and displaced volume ∇ refers to one of the hulls; the catamaran displacement then becomes $\Delta = 2.\rho.L.b.T.C_B$. The estimate of the overall depth D_O (including superstructure) in Table 9.1 is only approximate, and is provided primarily for use in the equipment numeral E for the hull and superstructure mass estimate.

As the principal hull parameters did not show any reliable trends with speed, the first estimate of dimensions in the iterative cycle is based only on passenger and vehicle requirements, together with appropriate values of hydrodynamic parameters as starting points. This creates an anomaly in the design procedure. For example, a change in speed for a particular design, whilst retaining the same passenger and vehicle requirements, results in a change in propulsive power and machinery mass and hence overall mass balance. This problem is overcome by incorporating a mass balance directly within the procedures for the derivation of dimensions, Figures 9.7 and 9.8.

In the design path, Figures 9.7 and 9.8, suitable values for $L/\nabla^{1/3}$, C_B and B/T are chosen and used in Equation (9.1). These may then be modified in further design iterations in order to achieve a satisfactory balance of masses, generally by adjusting the displacement. There are several ways in which the parameters may be modified, but an approach which has been found to be effective and efficient is to retain overall constancy of L/B, hence constant L from Equation (9.3), which results in constancy of Equation (9.1). Hence for constant L/B, combinations of $L/\nabla^{1/3}$, C_B and B/T within Equation (9.1) may be chosen depending on any other design constraints. For example, (i) for fixed ∇ and $L/\nabla^{1/3}$, C_B can be increased and B/T increased to retain constant ∇; (ii) if a change in ∇ is accepted, C_B and $L/\nabla^{1/3}$ may be changed with B/T constant, or B/T and $L/\nabla^{1/3}$ changed with C_B constant or suitable changes made to both B/T and C_B. The procedure for catamarans is similar, but using Equations (9.4) and (9.5).

It is seen that the proposed approach truly integrates areas and masses into the initial design

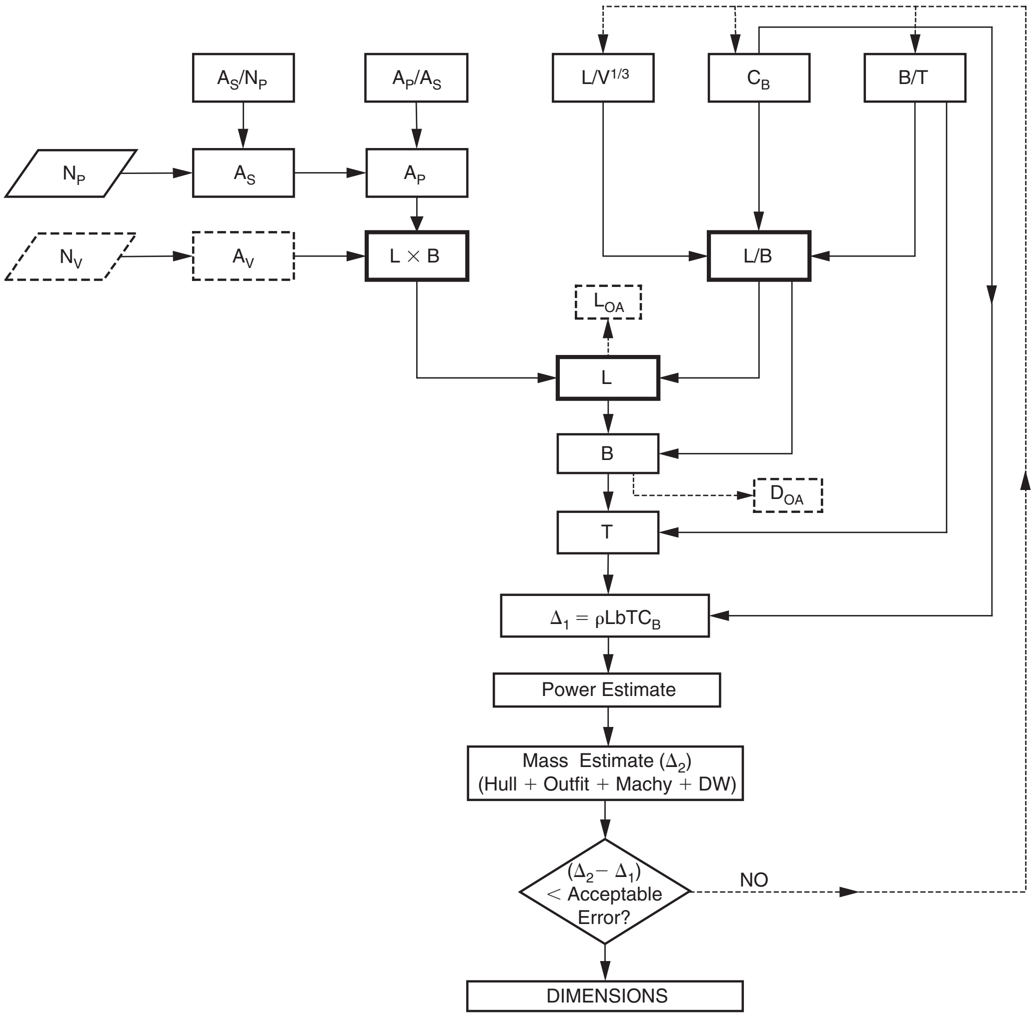


Figure 9.7 Estimation of main dimensions – monohulls.

cycle for the derivation of the dimensions. It is quite different from the more traditional approaches where, when establishing preliminary dimensions for a capacity carrier, the emphasis might be placed on volumes or areas followed by a mass check or, for displacement vessels, using a balance of masses followed by a capacity check.

A starting point in the design process can be established by using the mid values of the various parameters given in Table 9.1.

9.2.5 Stability check

Criterion for transverse stability: The transverse metacentric height GM may be used as a measure of ship stability, where GM may be calculated as:

$$GM = (KB + BM) - KG \tag{9.6}$$

(see also Chapter 3 for a more detailed discussion of stability)

KG — Depends on disposition of structure and contents; can be calculated in detail at advanced stage of design if time allows, or by inclining experiment after launch.

KB — Calculated from a knowledge of the underwater form.

BM — J_T/∇
J_T = transverse 2nd moment of area of the waterplane about the centreline.
∇ = immersed volume.

Estimation of GM at the preliminary design stage e.g. for use in the model described in Figure 9.5.

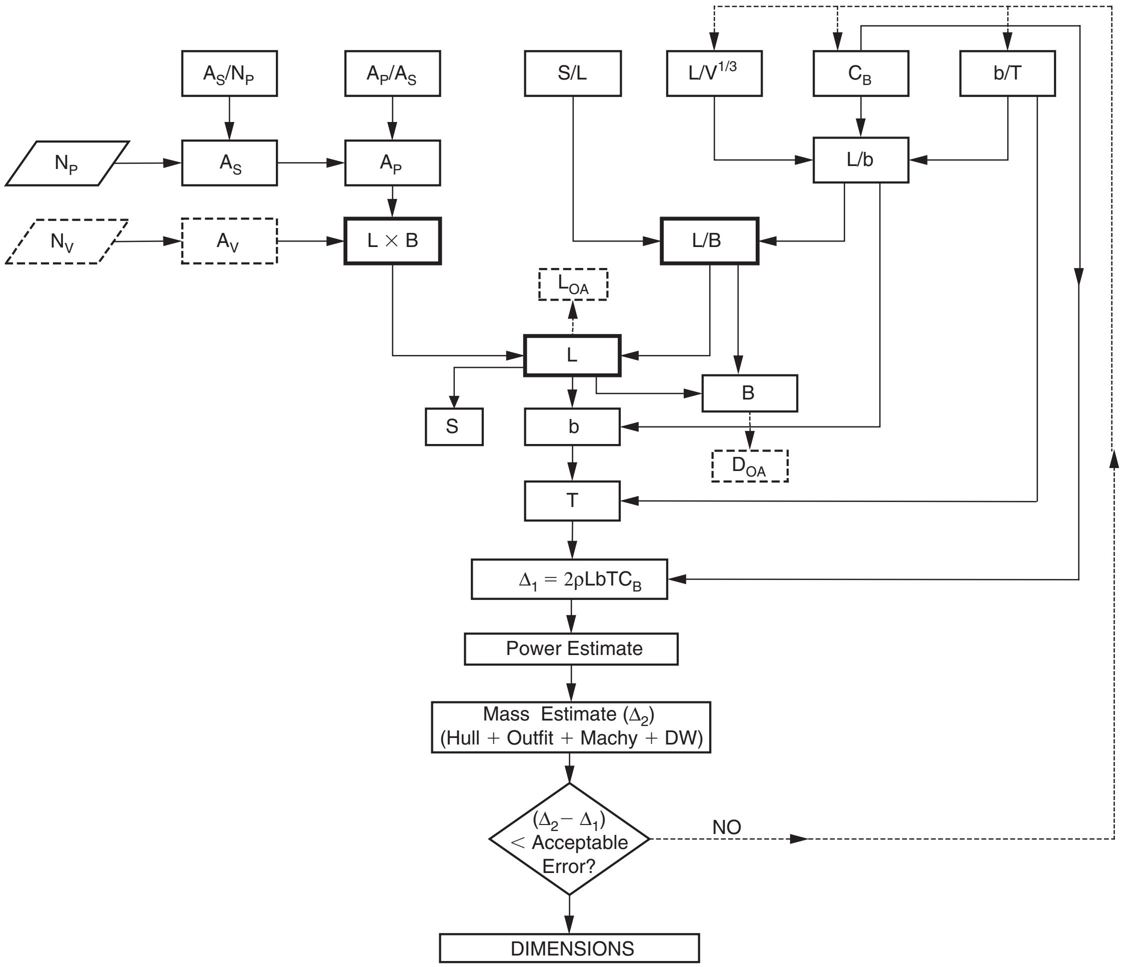


Figure 9.8 Estimation of main dimensions – catamarans.

Table 9.1 Design equations and range of parameters.

Item	*Pass. Only Monos*	*Pass. Only Cats*	*Car/Pass. Monos*	*Car/Pass. Cats*
$L \times B$ (m^2)	$146 + 1.86 \times 10^{-3} A_p^2$	$138 + 0.91A_p$	$121 + 0.27A_p + 0.60A_v$	$471 + 0.55A_p + 0.28A_v$
A_s/N_p (m^2)	0.55–0.75	0.55–0.85	0.85–1.25	0.80–1.40
A_p/A_s	1.10–1.30	1.10–1.30	1.15–1.45	1.30–1.70
A_v (m^2)	–	–	$156 + 10.2N_v$	$12.4N_v$
S/L	–	0.20–0.25	–	0.20–0.25
$L/\nabla^{1/3}$	5.0–7.5	8.0–10.5	6.5–9.0	8.5–11.0
(majority)	(5.5–6.5)	(8.5–9.5)	(7.0–8.5)	(9.5–10.5)
B/T	3.5–8.5	b/T 1.5–3.0	3.5–7.5	b/T 1.5–3.0
(majority)	(4.0–6.5)		(4.5–6.5)	
D_O	4 + 0.6B	4 + 0.44B	4 + 0.6B	4 + 0.44B
C_B	0.35–0.45	0.40–0.55	0.35–0.45	0.40–0.55
L_O/L	1.13–1.15	1.13–1.15	1.13–1.15	1.13–1.15

where A_P = Total pass. area (m^2), A_S = Seating area (m^2), A_V = Vehicle area, N_P = No. of pass. N_V = Number of vehicles

KG, KB and J_T will not be known accurately at the preliminary stage. In this case, empirical relationships can be used to provide satisfactory approximate estimates.

e.g. $KB = T/6[(5C_W - 2C_B)/C_W$ or
$KB = T[C_W/(C_W + C_B)]$

C_w = waterplane area coefficient = waterplane area/L × B

i.e. $KB = f_i(T)$
$BM = J_T/\nabla$
$\nabla = L.B.T.C_B$ hence known
$J_T = iLB^3$

i for waterplane shape can be related to C_W or C_B

Hence $BM = J_T/\nabla = f_2[LB^3/LBTC_B] = f_2[B^2/T\,C_B]$

KG is often defined as a function of depth D, i.e. $KG = f_3(D)$. For example, typical values for various ship types are as follows:

KG lightship	=0.63–0.7 D
	=0.69–0.66 D Tankers
	=0.63–0.66 D Bulk carrier
	=0.66–0.68 D Cargo
	=0.71–0.75 D Cargo Insulated
	=0.90 D Tug
	=0.84 D Trawler

KG loaded	=0.53 D Tankers
	=0.57 D Bulk carriers
	=0.65 D Container ships

At the preliminary stage it is normally satisfactory to assume f_1, f_2 and f_3 constant for a particular ship type. Assuming a value of $GM > 0.5\,m$ may be a typical criterion at the preliminary design stage. A maximum GM to preclude the possibility of a very small roll period (which may be undesirable, leading to high accelerations) may also be incorporated. Roll period for general cargo vessels can be approximated as:

$$P = \frac{0.43\,B}{\sqrt{GM}}\,secs \tag{9.7}$$

(i.e. large GM implies short periods and high accelerations) hence, if a minimum period of say 10 secs. is required, then $GM < 0.0018\ B^2$ is required.

Therefore a possible overall stability criterion/constraint for use in the model in Figure 9.5 might be:

$$0.0018\,B^2 > f_1(T) + f_2\left[\frac{B^2}{TC_B}\right] - f_3(D) > 0.5 \tag{9.8}$$

Note: There may be practical variations in GM due to distortion of dimensions, e.g. B limited for some Δ hence high L/B, low BM; shallow draught implies low KB; container ships with large deck cargo implies high KG etc.

9.2.6 Lightship mass estimates

The estimation of the masses of the various items which make up the lightship mass is an important factor in the design process. Masses have a bearing on the technical characteristics of the ship (such as draught and deadweight), and are often used as the basis for cost estimation see Section 9.6.4.2.

The lightship mass is normally summarized under three main headings:

1.	STEEL (W_s):	Steel hull and superstructure.
2.	OUTFIT (W_o):	Accommodation, deck fittings, piping, lifeboats etc.
3.	MACHINERY (W_m):	Main propulsion and auxiliaries such as generators, compressors, boilers etc.

A MARGIN will also be incorporated, depending on the level of uncertainty of the lightship estimate.

Mass estimates, both at the preliminary and detailed design stages are usually grouped under these headings.

It is possible at the detailed design stage, and particularly during and just after construction, to derive reasonably accurate estimates of these masses, although a lot of effort and time will usually be involved.

At the preliminary design stage rapid estimates are required and empirical approximations which relate the component masses to the principal ship particulars (such as dimensions and power) have to be used. The variables in the relationships usually have a physical justification, and the relationships will be 'calibrated' for different ship types. It is the duty of a naval architect to update such empirical relationships whenever possible.

9.2.6.1 *Steel mass*

This normally forms a significant part of the hull mass. Since total ship mass must equal displacement, fixed for a given vessel, a change in steel mass leads to a change in deadweight.

Steel mass/Δ_{mt} should be as low as possible with typical values as follows:

Cargo Ship	Steel/Δ%	20
Cargo & Passenger		28
Passenger		30
Cross Channel		35
Oil Tankers		18

Factors affecting steel mass include: draught (in relation to dimensions), proportions (e.g. L/D), fineness or fullness of form, number of decks and bulkheads, extent of deckhouses and erections, type of construction (structural design).

Use of recorded steel mass for a basis ship forms the most common method of making a preliminary estimate for a new proposal.

Care is required as to what basis steel mass includes: i.e. some yards include hull forgings and castings. Further, whilst generalized data gives a good guide, care must be exercised in interpretation as even for similar ships (or even sisters) steel masses can differ due to alternative methods of construction, owners extras, classification requirements for special vessels/conditions etc.

For *special ship types*, or *novel designs*, detailed mass calculations based on preliminary plans may have to be resorted to.

Steel *ordered* is subject to a rolling margin of ±2½%. Steel purchased is *invoiced mass* and hence steel received varies to within ±2½% of that ordered.

Steel built into the ship is known as *Net Steel Mass*; net steel is about 8–10% less than invoiced mass, i.e. 8–10% scrap.

Methods of estimating steel mass:

(a) Cubic number:

Used for preliminary estimates only.

$$C_N = \frac{LBD}{1000} \text{ where D is to uppermost deck}$$

$$\text{and steel mass } (W_S) = C \cdot \frac{LBD}{1000},$$

where C is some constant derived from basis vessel(s). Method attaches no importance to draught or erections. It also assumes L, B and D to influence the steel mass by the same amount which is not true. Such an approach has to be used with caution. Only if the basis ship is similar and there is little difference in L, B and D can good results be obtained.

(b) Dimensional corrections and differences:

This approach is normally based on data for a basis vessel. Dimensional corrections can be made for length, breadth and depth separately, with subsequent allowances for any other differences between basis and design.

For the dimensional correction, it is required to have the mass/unit change in length, breadth and depth. Also, since steel mass is more sensitive to some dimensions than others it is assumed, for example, that of steel mass:

85% is affected by L
55% is affected by B
30% is affected by D.

This increase in steel mass can be written as follows:

$$\text{increase} \quad \delta W_S = 0.85\, w_1 \left[\frac{L_2 - L_1}{L_1}\right] + 0.55\, w_1 \left[\frac{B_2 - B_1}{B_1}\right] + 0.30\, w_1 \left[\frac{D_2 - D_1}{D_1}\right] \quad (9.9)$$

where w_1 = steel mass of basis.

As well as the basic *dimensional* correction, *difference* corrections will also be made for changes in scantlings due to change in dimensions, change in *form*, *sheer*, and any other changes such as in erections, superstructures, bulkheads etc., Munro-Smith (1950).

Corrections for changes in sheer (which will normally be small) erections, superstructures and watertight bulkheads etc. would then be carried out as required.

It should be noted that, if sufficient mass data are available for vessels of similar type, the 'weightings' or importance of the various dimensions can be derived as follows:

$$\text{assume} \quad W_s = k\, L^a\, B^b\, D^c \quad (9.10)$$

taking logs

$$\log W_s = \log k + a \log L + b \operatorname{Log} B + c \operatorname{Log} D \quad (9.11)$$

$$\text{differentiate:} \quad \frac{dW_S}{W_S} = a\frac{dL}{L} + b\frac{dB}{B} + c\frac{dD}{D}$$

i.e. in same format as Equation (9.9), and where coefficients a, b and c may be obtained from multiple linear regression of Equation (9.11). Alternatively, once the coefficients are determined, Equation (9.10) may be used directly.

C_B, T and other variables may be added to Equation (9.10) provided adequate parametric data for similar vessels are available.

(b) Mass/Unit length:

(i) Method uses the midship section for both basis *and* new design. Steel mass for new design

proportioned on the change in mass/m and change in length.
i.e. if W = mass of steel for basis, W_1 = mass/m for basis and W_2 = mass/m for new design, then:

$$\text{Steel mass for new design} = W \times \frac{W_2}{W_1} \times \frac{L_2}{L_1}. \quad (9.12)$$

The method assumes the mass for each ship to be distributed in the same proportion to each other throughout length as they do at amidships. Further corrections may be made, as necessary, for changes in erections, bulkheads etc.

(ii) Use may also be made of mass/m at the preliminary design stage (without a basis vessel) by estimating the mass/m amidships (from midship section) using Classification Society rules, and distributing mass through ship say according to Sectional Area Curve (see proposal by Watson and Gilfillian (1977)). Integration of the mass distribution will give total mass. The method has seen more applications in recent years, since the Classification Rules are available on computer. Parametric variation of dimensions allows a database to be established (for a particular vessel type) and regression equations may be fitted to the data for design purposes.

(c) Group mass method:
Steel mass of known ships analysed into suitable subdivisions or groups. For each group a parameter proportional to say volume or area is derived which can be applied to new designs. Typical groups will include: shell plating, framing, bulkheads, deck plating, erections etc.

The method is best suited to shipyards who have detailed data, and who have established computer data bases, normally also including the hours to work the materials in a particular group. Such an approach allows total masses, building costs and scheduling to be estimated.

(d) Steel mass as function of Lloyds equipment numeral:
Proposed by Watson (1962), updated by Watson and Gilfillian (1977).

Net steel mass plotted against Lloyds equipment numeral:

$$E = L(B + T) + 0.85\,L\,(D - T) + 0.85\,\Sigma l_1 h_1 + 0.75\,\Sigma l_1 h_2$$

where l_1 and h_1 = length and height of full width erections
l_2 and h_2 = length and height of houses.

If extent of houses/erections not known at design stage, for ordinary cargo ships an allowance of 200–300 (metric units) can be used.

(Numeral shown is in fact 'old' numeral, and in 1962 paper Watson plotted invoiced steel – 1977 paper retains 'old' numeral but plots net steel).

Steel masses plotted by Watson were corrected to standard fullness C_B = 0.7, measured at 0.8D.

Corrections to steel mass for variation in C_B from 0.7 are made using the following relationship:

$$W_S = W_{S_{0.7}}(1 + 0.5(C_B' - 0.7)) \quad (9.13)$$
W_S = steel mass for actual C_B' at 0.8D

$W_{S_{0.7}}$ = steel mass at C_B' = 0.7 as lifted from graph (or following equation).

Watson found following formula to give satisfactory fit to data:

$$W_{S_{0.7}} = kE^{1.36} \quad (9.14)$$

with k for different ship types as shown in Table 9.2:

Table 9.2 k values for steel mass.

Type	*k*
Tankers/bulk carriers	0.029–0.035
Containers	0.033–0.040
Cargo	0.029–0.037
Tugs	0.044
Trawlers	0.041–0.042
Ferries	0.024–0.037
Passenger	0.037–0.038

Hence combining block coefficient correction with above formula:

$$\text{Net steel mass} = kE^{1.36}\,(1 + 0.5(C_B' - 0.7)) \quad (9.15)$$

This method offers a good approach at the preliminary design stage for the relevant ship types.

(e) Detailed (direct) calculations:
These are lengthy and laborious. They are required for unusual design proposals. Method does yield LCG, VCG. Ship has to be fairly well defined with approximate body plan, position of DB, DKs, 1/2 girths, frame spacing and outline of section and scantlings etc.

9.2.6.2 *Outfit mass*

(i) Dimensional corrections:
This approach is normally based on data for a basis vessel. It assumes that part of the outfit mass is constant between basis and new design and the

remainder to vary as length and breadth. Hence the corrected mass for new design W_{O2} is given by:

$$W_{O2} = xW_{O1} + (1-x)\left[W_{O1} \times \frac{L_2}{L_1} \times \frac{B_2}{B_1}\right] \quad (9.16)$$

where the value of x will depend on ship type and size, say 0.5 in the absence of better information.

The approach will normally be too inaccurate for vessels such as ferries and passenger vessels etc.

(ii) Empirical approach:
Uses typical empirical formulae for outfit mass based on L × B for various ship types. Values proposed in the Watson and Gilfillian (1977) paper are as follows:

$$W_O = k' \times L \times B \text{ tonnes}$$

With typical values for k′ shown in Table 9.3:

Table 9.3 k′ values for outfit mass.

Passenger ships	K′ = 0.7–1.55 (L = 100–250m)	Intermediate values by linear interpolation
Trawlers	k′ = 0.3–0.5 (L = 25–80m)	
Cargo vessels	k′ = 0.4	
Container	k′ = 0.32	
Tankers/Bulk Carriers	k′ = 0.25–0.18 ….(L = 150–300m)	

Such formulae and plotting of empirical data can be very approximate due to wide variations in outfit mass that can occur for a particular ship type. They must be used with care.

9.2.6.3 *Machinery mass*

It is important to note that the *TOTAL* machinery mass is made up of the main propulsion machinery *together with* the remaining machinery such as auxiliaries, compressors, boilers, piping etc.

A preliminary power estimate is required prior to carrying out the machinery mass estimate. This may be made using standard series or suitable regression data or simple relationships based on displacement and speed (e.g. Power $\alpha\ \Delta^{2/3}V^3$, see Chapter 5).

In estimating the machinery mass the most effective approach is to break down the total machinery mass into the propulsion machinery mass and the remainder. Data is readily available for main engine(s), allowing data fits to be made and updated. Such an approach, and the equations proposed by Watson (1977) is as follows:

For diesel installation:

$$\text{Main engine(s) mass} = 9.38\left[\frac{P}{N}\right]^{0.84} \quad (9.17)$$

where P = installed power (HP) and N = engine rpm (not prop) (a typical assumption at the preliminary design stage is N = 110 rpm for low speed diesel and 500 rpm for medium speed diesel).

Remaining mass (tonnes) = $k'' P^{0.7}$

where k" = 0.56 for bulk carriers and general cargo
= 0.59 for tankers
= 0.65 for passenger vessels and ferries

Hence, TOTAL Machinery mass W_m (tonnes)

$$= 9.38\left[\frac{P}{N}\right]^{0.84} + k'' P^{0.7} \quad (9.18)$$

For a steam turbine installation, Watson proposes:

$$\text{Total machinery Mass tonnes} = 0.16\,(\text{SHP})^{0.89}$$

where SHP = installed maximum power.

9.2.6.4 *Margin*

The sum of the net steel and outfit constitute the hull mass; any underestimation can only be made up for in loss of deadweight, DW. Also any departures from design causing increase in hull mass will influence DW. Thus a margin is normally allowed. Amount of margin allowed will depend on the degree of uncertainty of the lightship estimate, and penalty clauses regarding non-compliance with the specified deadweight.

Typical values are:

1% of Lightship mass + 0.1% load displacement	Watson (1962)
2% of DW	Munro-Smith (1950)
2% Lightweight	Watson and Gilfillian (1977)

Such margins normally adjusted to give a round figure for the lightship mass.

9.2.6.5 *Masses of fast ferries*

Estimation procedures for the masses of the aluminium alloy hull, outfit and machinery for fast ferries (monohulls and catamarans) have been proposed by Karayannis *et al.* (1999), Karayannis and Molland (2001) and Molland *et al.* (2003) using methods similar to those already described. Satisfactory values were obtained when compared with data from basis ships, and the methods are particularly suitable for use at the preliminary design stage.

9.2.6.6 *Vertical centre of gravity (KG)*

A *detailed* mass check will normally incorporate VCG and LCG information.
Actual ship KG can be derived from an inclining experiment, see Section 3.6.

For approximate and preliminary purposes (such as for stability check in Section 9.2.5), KG normally expressed as a function of depth (D) for a particular ship type/size. This function may be derived from a basis vessel (correcting for depth and changes in machinery mass KG etc and any other significant changes) or a database of similar vessels, and applied to a new design. For cargo ships, bulk carriers etc lightship KG is typically 0.68 to 0.72D.

If more detailed data are available, different levels of breakdown of the components of KG may be applied.

9.2.7 Design of ship lines

The design of the ship's lines is fundamental to the ship design process. This Section considers the design of the ship lines and body plan (see Figures 3.2 and 9.44), and the modification to form.

There are several ways to establish the lines and body plan but four fundamentals must be achieved:

1. Correct displacement Δ on selected principal dimensions.
2. Correct LCB – determined partly by disposition of structure/cargo/machinery, etc (LCG) and also by best form for resistance.
3. Position of metacentre in worst condition of loading of vessel – dependent on beam and shape of waterline.
4. Shape of Sectional Area Curve (SAC) for satisfactory propulsion; Cp, entrance, run, LCB and maximum slope of SAC.

Two distinct phases in designing the lines:

(a) Achievement of form characteristics, per 1–4 above.
(b) Ensuring form determined corresponds to a fair body.

- The body plan is normally drawn to moulded lines. Thus form coefficients for steel ships are normally for moulded displacement; a typical allowance for the shell is 5 tonnes/1000 tonnes displacement (0.5%) to give *extreme* displacement.
- A body plan can be developed from first principles. However, a similar basis ship, or suitable published body plan is frequently chosen and modified to the correct C_B, C_p, LCB, L, B, etc for the new design.
- Changes to the basis for the proposed design, such as to C_p and LCB, are most conveniently carried out by modifying the basis Sectional Area Curve (SAC). It is a very robust method and allows complete control over any changes. The Sectional Area Curve will also be employed when developing a body plan/lines from first principles.

9.2.7.1 *Sectional area curve (SAC) – definitions:*

$\bar{x}$ = LCB = longitudinal centre of area of curve

$$\frac{V}{L.A_m} = Cp_T = \frac{\text{Area Under Curve}}{\text{Enclosing Rectangle}}$$

$$Cp_F = \frac{\text{Immersed vol. for 'd Amidships}}{L/2 \times A_m}$$

$$Cp_E = \frac{\text{Immersed vol. of Entrance}}{E \times \text{Largest section area}}$$

Cp_A, Cp_R obtained similarly

It is usual to normalise the length in terms of stations 0∼10 (or 0∼1) and nondimensionalise areas in terms of Area amidships A_m, Figure 9.10.

Using this notation, C_P equals the area under the sectional area curve

9.2.7.2 *Modifications to sectional area curve*

(a) To change Cp:

$$\frac{r_1}{r_2} = \frac{1 - Cp'}{1 - Cp}$$

Cp′ = desired Cp for new design (correction applied both ends), Figure 9.11.

Modifying Cp in this way also changes LCB, hence logical procedure is to

(a) correct for Cp, and recalculate LCB
(b) correct to LCB of new form.

(b) To change LCB:
Find the centre of area of SAC (longitudinal and about base), and modify as shown in Figure 9.12.

$$\delta x(\%L) = \frac{BB'(\%L)}{\bar{y}} \times y$$

The simple (1 − Cp) change as described above has a number of disadvantages:

(1) There is no control over Parallel mid body, PMB, in the derived form
(2) It cannot be used to reduce Cp in a form with no PMB

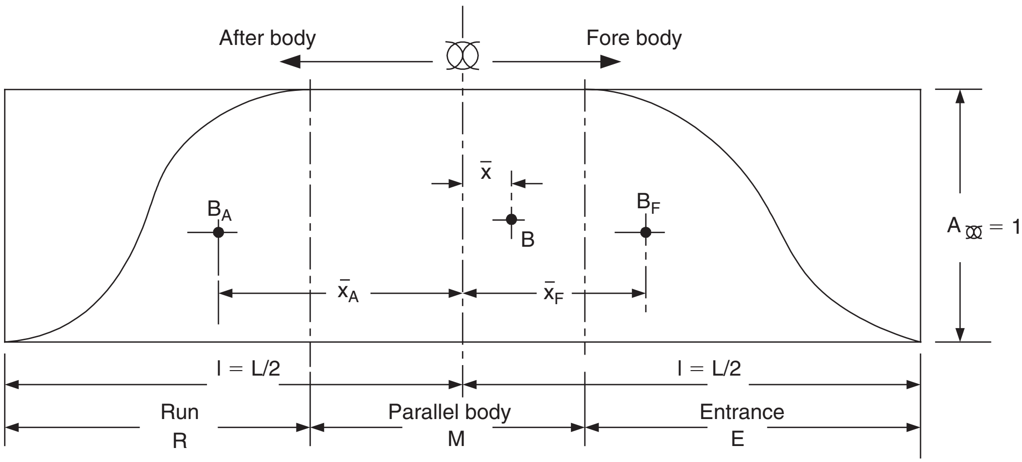


Figure 9.9 Sectional area curve.

(3) The basic form without PMB cannot be increased in fullness without inserting PMB
(4) The prismatic of entrance and run cannot be adjusted.

The above simple methods are however suitable for many applications.

These deficiencies can be overcome using the Lackenby Transformations, described in the next section.

9.2.7.3 *Sectional area curve transformations*

Useful procedures are the Lackenby transformations, Lackenby (1950).

The deficiencies in the $(1 - Cp)$ and LCB curve swinging methods can be overcome by the more detailed numerical calculations described by Lackenby.

Lackenby includes the basic $(1 - Cp)$ and curve swinging methods and makes a comprehensive review of alternatives including the cases of:

$(1 - Cp)$ method } either holding LCB constant or changing LCB

Change in Cp for case of no parallel mid body } either holding LCB constant or changing LCB

- *Summary of formulae*: which are applied to forward and aft *separately*

ϕ = original Cp, $\delta\phi$ = change in Cp see Figure 9.13

- One-minus Prismatic method

$$\delta x = \frac{\delta\phi}{(1-\phi)}(1-x), \quad \text{i.e. same as } r_2, r_1 \text{ method.}$$

also $\delta p = \dfrac{\delta\phi}{(1-\phi)}(1-x)$, where P = original PMB.

$$h = \frac{\phi(1-2\bar{x})}{(1-\phi)}$$

$\bar{x}$ = centroid of original area – forward or aft

h = centroid of added 'sliver'

(h is used in derivation of $\delta\phi_f$ or $\delta\phi_a$)

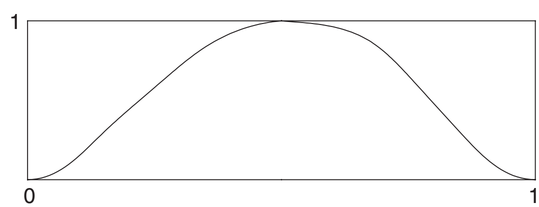


Figure 9.10 Sectional area curve.

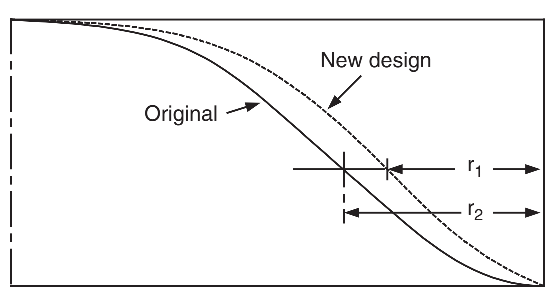


Figure 9.11 SAC: Change in Cp.

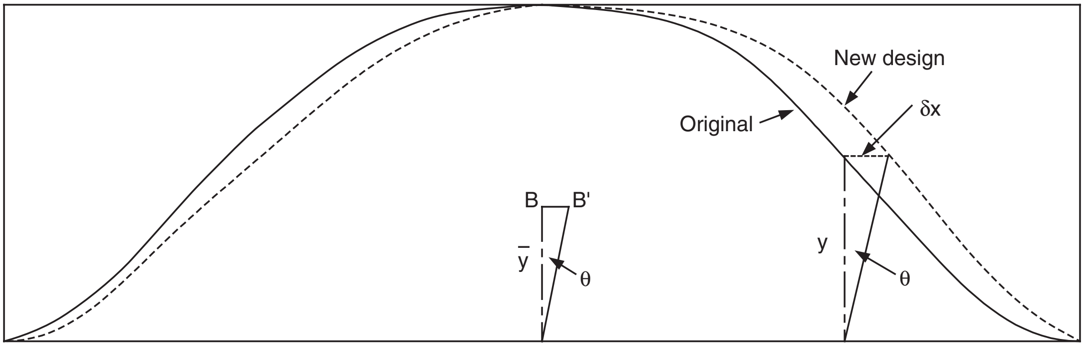


Figure 9.12 SAC: Change in LCB.

- Case with no PMB:
  δx variation assumed as $\delta x = c.x(1 - x)$
  i.e. maximum change at $x = 0.5$. (max. shift restricted to shoulders in the case of (1 − Cp) variation).

whence $\delta x = \dfrac{\delta\phi x(1-x)}{\phi(1-2\bar{x})}$ and $h = \dfrac{2\bar{x} - 3k^2}{(1-2\bar{x})}$

where k = lever of second moment of original curve, Figure 9.13.

- Derivation of $\delta\phi_f$ and $\delta\phi_a$ to suit required LCB change (or need to keep constant).

  [Suffixes f and a represent forward or aft].

$$\delta\phi_f = \frac{2[\delta\phi_t\,(h_a + \bar{z}) + \delta\bar{z}\,(\phi_t + \delta\phi_t)]}{(h_f + h_a)}$$

and $$\delta\phi_a = \frac{2[\delta\phi_t\,(h_f - \bar{z}) - \delta\bar{z}\,(\phi_t + \delta\phi_t)]}{(h_f + h_a)}$$

ϕ_t = total Cp
$\bar{z}$ = distance of LCB of Basis ship from amidships, as a fraction of half-length (+ve forward)
$\delta\bar{z}$ = required fractional shift of LCB.

also: $\phi_t = (\phi_f + \phi_a)/2$ and $\delta\phi_t = (\delta\phi_f + \delta\phi_a)/2$

Examples of special cases:
e.g. if LCB is to remain unchanged, $\delta\bar{z} = 0$

and $\delta\phi_f = \dfrac{2\delta\phi_t(h_a + \bar{z})}{(h_f + h_a)}$; $\delta\phi_a = \dfrac{2\delta\phi_t(h_f - \bar{z})}{(h_f + h_a)}$

or if ϕ_t is to remain unchanged, $\delta\phi_t = 0$

and $\delta\phi_f = \dfrac{2\delta\bar{z}.\phi_t}{(h_f + h_a)}$ and $\delta\phi_a = \dfrac{-2\delta\bar{z}\phi_t}{(h_f + h_a)}$

- Proofs of the above formulae are given in Lackenby's paper and applications are illustrated by worked examples.

9.2.7.4 *Preparation of body plan*

(A) From '*FIRST PRINCIPLES*': Using Sectional Area and load water line curves:

Midship Area:
There is not much freedom with midship area coefficient (C_m). There are high values in cargo ships/tankers leading to large cargo carrying capacity.

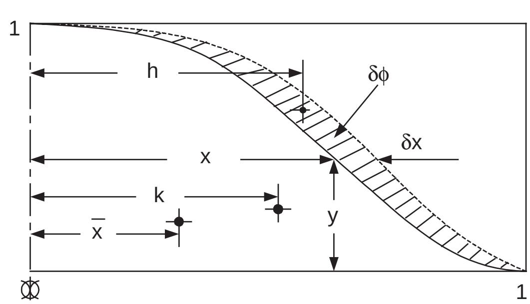


Figure 9.13 Notation for Lackenby transformation.

Typical values:

C_B	.75	.70	.65	.6	.55	$C_B = Cm.Cp$
Cm	.987	.984	.980	.975	.960	

i.e. as C_B reduces, C_m reduces.

In fine form ships low C_m implies large rise of floor and large bilge radius.

Choice of rise of floor may also depend on directional stability and drainage from double bottom tanks. Bilge radius varies with fullness of section C_m.

Section Design: Assuming Sectional Area Curve and Waterline are available, e.g. using a polynominal approach, or from standard series data.

For a particular station, area A from SAC, B/2 from LWL curve hence proceed as shown in Figure 9.14.

Repeat at other sections, and fair using waterlines.

(B) Body plan '*FROM BASIS*': with change in Cp and/or LCB.

Correct the basis SAC and lift offsets for new design from basis ship lines at a revised station spacing from ends, Figure 9.15. New lines are automatically fair.

Trans immersed area at 9A = area at 9′E

At position 9′ on basis lines (1/2 breadth) plan, lift waterline offsets and plot at station 9 for new design; repeat for other stations. This procedure results in fair new lines and body plan.

Correct for $\frac{T_1}{T}$ and $\frac{B_1}{B}$ if necessary (maintains C_B and LCB).

Note: this method (B) lends itself to a computer based approach.

i.e.

Digitize basis body plan
↓
Derive sectional area curve, Cp, LCB for BASIS
↓
Swing/modify SA curve to suit new design, and obtain shift of stations δx
↓
Cubic spline WL curves put through BASIS data
↓
Interpolate basis WLs at new stations (orig. $\pm\ \delta x$) } Check new Cp, LCB
↓
Draw body plan for new design

Alternatively, the required shift of stations may be applied to commercial ship lines packages.

(C) *STANDARD SERIES DATA*:

A number of standard series have been published which provide a useful source of hull forms, as well as providing resistance/powering data.

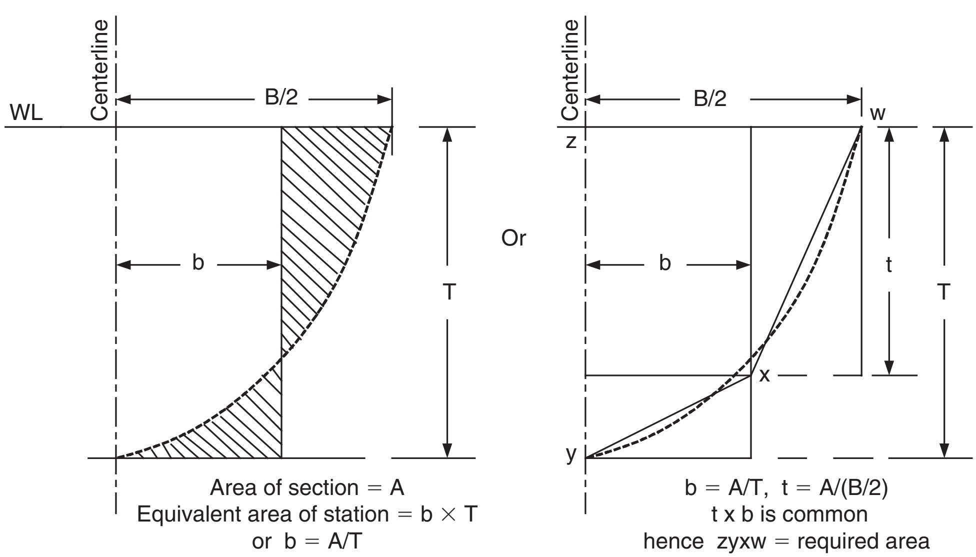


Figure 9.14 Section design.

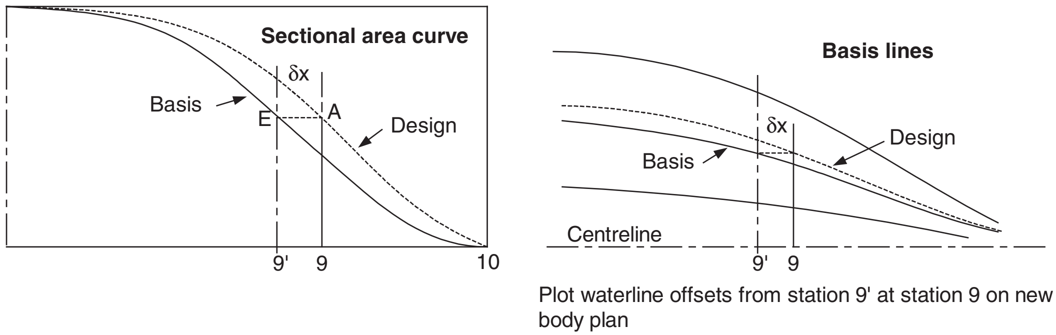


Figure 9.15 Modifying ship lines.

BSRA Series:	*Trans. RINA* 1961, 1966 BSRA Report No. NS 333 gives refaired lines + bulbous bows	SS Merchant Forms
Dawson Coasters:	*IESS* 1958/59 & earlier	
Series 60:	Todd, *SNAME* Vol. 61, 1953	
Taylor Series:	Taylor/Gertler revised DTMB Rep. 806	TS
Linblad Series:	*Trans RINA* 1946/49	
NPL Round Bilge Series:	Smaller Semi-Displacement Craft RINA Monograph No. 4	
Series 64 Round Bilge Series:	*Marine Technology*, No. 2, July 1965, SNAME	
Series 62 Planing Hulls:		
NTUA Series (double chine):	Semi-displacement craft. Radojcic *et al.* (2001)	

See also Section 5.1, Resistance and propulsion.

9.2.8 Statutory regulations

Legislation exists which is concerned with the safety of ships and the well being of all who sail in them. International legislation with regard to shipping is now dealt with by the I.M.O. (International Maritime Organization, formerly I.M.C.O.) which was set up in 1959 by the U.N. Its various committees meet periodically and I.M.O. arranges various conferences such as SOLAS 1960, 1974, International Load Line Conference 1966 and the Tonnage Conference 1969. Further description and discussion of the role of IMO is included in Chapter 11.

Implementation of the legislation is the responsibility of the government of the country concerned. In the UK it is administered by the Maritime and Coastguard Agency (MCA) and the rules are drawn up by virtue of the Merchant Shipping Acts.
Surveyors verify that ships are built and operated in accordance with the regulations.

Typical matters (and hence legislation) with which the rules are concerned are: Stability; Load lines (freeboard); Subdivision; Tonnage; Life Saving Appliances; Crew Accommodation Regulations; Fire Appliances and Protection; Carriage of Grain Cargoes; Dangerous Cargoes.

Since the regulations are statutory they are of fundamental importance in the design and operation of ships and, consequently, have to be integrated in the design process from the early conceptual stages to the detailed final stages. Stability, freeboard and subdivision, in particular, are fundamental to the initial design process. Stability and subdivision are described and discussed in Chapter 3. Information on freeboard and tonnage may be obtained from Eyres (2007), Tupper (2004), IMO (1966) and IMO (1969).

9.2.9 Concept design content: example

The typical content which may be covered in the design process at concept stage is shown in Figure 9.16, and applies much of the content in the earlier Sections of this Chapter. Aspects such as powering, structures, seakeeping and manoeuvring and safety are dealt with in other Chapters.

9.3 Materials

9.3.1 Introduction

A description is given of the principal materials used in the construction of the main components of a

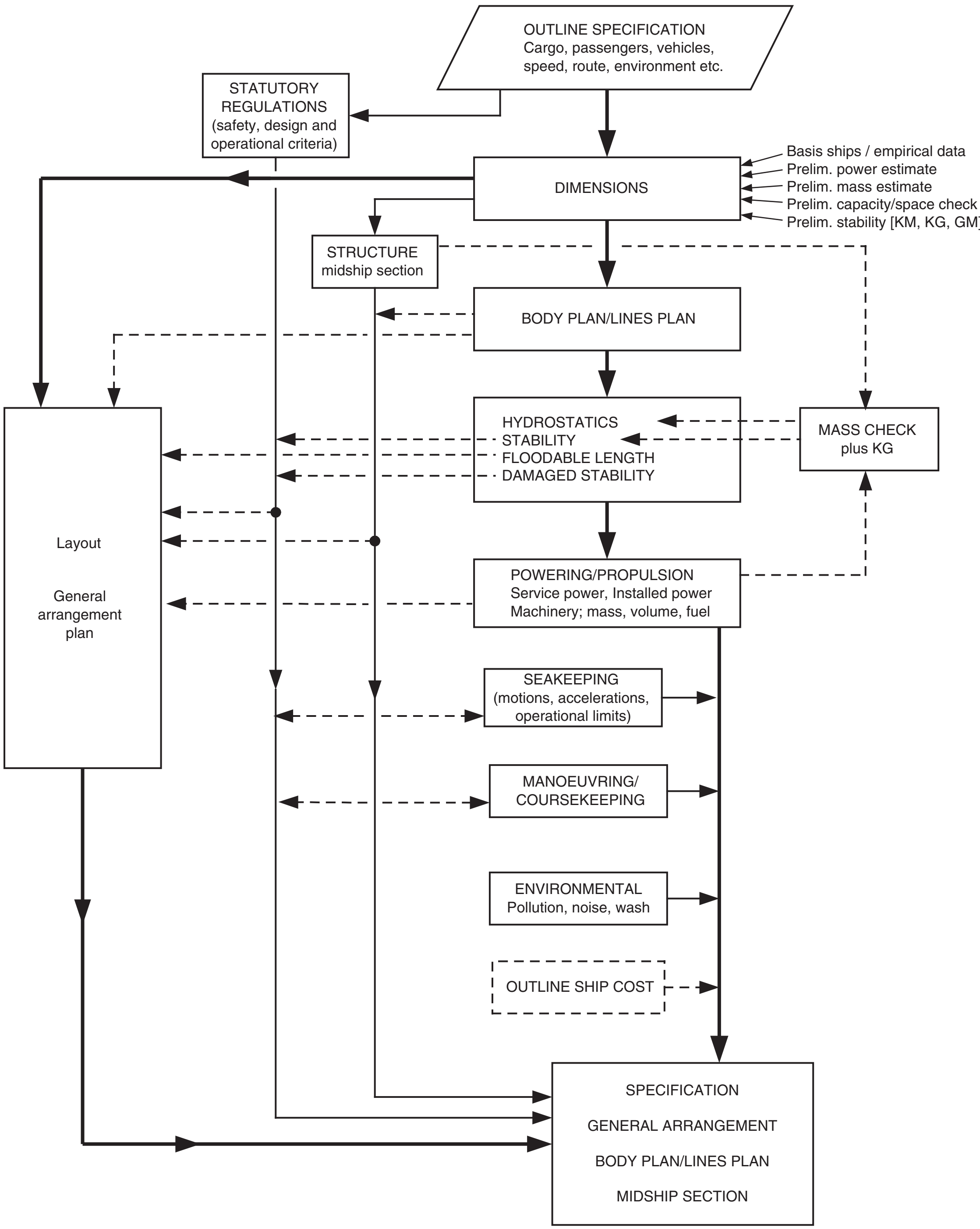


Figure 9.16 Typical content of concept design model.

ship or marine structure. These amount basically to steels, aluminium alloys and composites. An outline of corrosion, corrosion control and anti-fouling is included.

9.3.2 Steel

The production of all steels used for shipbuilding purposes starts with the smelting of iron ore and the making of pig-iron. Normally the iron ore is smelted in a blast furnace, which is a large, slightly conical structure lined with a refractory material. To provide the heat for smelting, coke is used and limestone is also added. This makes the slag formed by the incombustible impurities in the iron ore fluid, so that it can be drawn off. Air necessary for combustion is blown in through a ring of holes near the bottom, and the coke, ore, and limestone are charged into the top of the furnace in rotation. Molten metal may be drawn off at intervals from a hole or spout at the bottom of the furnace and run into moulds formed in a bed of sand or into metal moulds.

The resultant pig-iron is from 92 to 97% iron, the remainder being carbon, silicon, manganese, sulphur, and phosphorus. In the subsequent manufacture of steels the pig iron is refined, in other words the impurities are reduced.

9.3.2.1 *Manufacture of steel*

Steels may be broadly considered as alloys of iron and carbon, the carbon percentage varying from about 0.1% in mild steels to about 1.8% in some hardened steels. These may be produced by one of four different processes, the open hearth process, the Bessemer converter process, the electric furnace process, or an oxygen process. Processes may be either an acid or basic process according to the chemical nature of the slag produced. Acid processes are used to refine pig-iron low in phosphorus and sulphur which are rich in silicon and therefore produce an acid slag. The furnace lining is constructed of an acid material so that it will prevent a reaction with the slag. A basic process is used to refine pig-iron that is rich in phosphorus and low in silicon. Phosphorus can be removed only by introducing a large amount of lime, which produces a basic slag. The furnace lining must then be of a basic refractory to prevent a reaction with the slag. About 85% of all steel produced in Britain is of the *basic* type, and with modern techniques is almost as good as the *acid* steels produced with superior ores.

Only the open hearth, electric furnace, and oxygen processes are described here as the Bessemer converter process is not used for shipbuilding steels.

Open hearth process. The open hearth furnace is capable of producing large quantities of steel, handling 150 to 300 tonnes in a single melt. It consists of a shallow bath, roofed in, and set above two brick-lined heating chambers. At the ends are openings for heated air and fuel (gas or oil) to be introduced into the furnace. Also these permit the escape of the burned gas which is used for heating the air and fuel. Every twenty minutes or so the flow of air and fuel is reversed.

In this process a mixture of pig-iron and steel scrap is melted in the furnace, carbon and the impurities being oxidized. Oxidization is produced by the oxygen present in the iron oxide of the pig-iron. Subsequently carbon, manganese, and other elements are added to eliminate iron oxides and give the required chemical composition.

Electric furnaces. Electric furnaces are generally of two types, the arc furnace and the high-frequency induction furnace. The former is used for refining a charge to give the required composition, whereas the latter may only be used for melting down a charge whose composition is similar to that finally required. For this reason only the arc furnace is considered in any detail. In an arc furnace melting is produced by striking an arc between electrodes suspended from the roof of the furnace and the charge itself in the hearth of the furnace. A charge consists of pig-iron and steel scrap and the process enables consistent results to be obtained and the final composition of the steel can be accurately controlled.

Electric furnace processes are often used for the production of high-grade alloy steels.

Oxygen process. This is a modern steelmaking process by which a molten charge of pig-iron and steel scrap with alloying elements is contained in a basic lined converter. A jet of high purity gaseous oxygen is then directed onto the surface of the liquid metal in order to refine it.

Steel from the open hearth or electric furnace is tapped into large ladles and poured into ingot moulds. It is allowed to cool in these moulds, until it becomes reasonably solidified permitting it to be transferred to ‘soaking pit’ where the ingot is reheated to the required temperature for rolling.

Chemical additions to steels. Additions of chemical elements to steels during the above processes serve several purposes. They may be used to deoxidize the metal, to remove impurities and bring them out into the slag, and finally to bring about the desired composition.

The amount of deoxidizing elements added determines whether the steels are 'rimmed steels' or 'killed steels'. Rimmed steels are produced when only small additions of deoxidizing material are added to the molten metal. Only those steels having less than 0.2% carbon and less than 0.6% manganese can be rimmed. Owing to the absence of deoxidizing material, the oxygen in the steel combines with the carbon and other gases present and a large volume of gas is liberated. So long as the metal is molten the gas passes upwards through the molten metal. When solidification takes place in ingot form, initially from the sides and bottom and then across the top, the gasses can no longer leave the metal. In the central portion of the ingot a large quantity of gas is trapped with the result that the core of the rimmed ingot is a mass of blow holes. Normally the hot rolling of the ingot into thin sheet is sufficient to weld the surfaces of the blow holes together, but this material is unsuitable for thicker plate.

The term 'killed' steel indicates that the metal has solidified in the ingot mould with little or no evolution of gas. This has been prevented by the addition of sufficient quantities of deoxidizing material, normally silicon or aluminium. Steel of this type has a high degree of chemical homogeneity, and killed steels are superior to rimmed steels. Where the process of deoxidation is only partially carried out by restricting the amount of deoxidizing material a 'semi-killed' steel is produced.

In the ingot mould the steel gradually solidifies from the sides and base as mentioned previously. The melting points of impurities like sulphides and phosphides in the steel are lower than that of the pure metal and these will tend to separate out and collect towards the centre and top of the ingot which is the last to solidify. This forms what is known as the 'segregate' in way of the noticeable contraction at the top of the ingot. Owing to the high concentration of impurities at this point this portion of the ingot is often discarded prior to rolling plate and sections.

9.3.2.2 *Heat treatment of steels*

The properties of steels may be altered greatly by the heat treatment to which the steel is subsequently subjected. These heat treatments bring about a change in the mechanical properties principally by modifying the steel's structure. Those heat treatments which concern shipbuilding materials are described.

Annealing. This consists of heating the steel at a slow rate to a temperature of say 850°C to 950°C, and then cooling it in the furnace at a very slow rate. The objects of annealing are to relieve any internal stresses, to soften the steel, or to bring the steel to a condition suitable for a subsequent heat treatment.

Normalizing. This is carried out by heating the steel slowly to a temperature similar to that for annealing and allowing it to cool in air. The resulting faster cooling rate produces a harder stronger steel than annealing, and also refines the grain size.

Quenching (or hardening). Steel is heated to temperatures similar to that for annealing and normalizing, and then quenched in water or oil. The fast cooling rate produces a very hard structure with a higher tensile strength.

Tempering. Quenched steels may be further heated to a temperature somewhat between atmospheric and 680°C, and some alloy steels are then cooled fairly rapidly by quenching in oil or water. The object of this treatment is to relieve the severe internal stresses produced by the original hardening process and to make the material less brittle but retain the higher tensile stress.

Stress relieving. To relieve internal stresses the temperature of the steel may be raised so that no structural change of the material occurs and then it may be slowly cooled.

9.3.2.3 *Steel sections*

A range of steel sections are rolled hot from ingots. The more common types associated with shipbuilding are shown in Figure 9.17. It is preferable to limit the sections required for shipbuilding to those readily available, that is the standard types; otherwise the steel mill is required to set up rolls for a small amount of material which is not very economic.

9.3.2.4 *Shipbuilding steels*

Steel for hull construction purposes is usually mild steel containing 0.15 to 0.23% carbon, and a reasonably high manganese content. Both sulphur and phosphorus in the mild steel are kept to a minimum (less than 0.05%). Higher concentrations of both are detrimental to the welding properties of the steel, and cracks can develop during the rolling process if the sulphur content is high.

Steel for a ship classed with Lloyds Register is produced by an approved manufacturer, and inspection and prescribed tests are carried out at the steel mill before dispatch. All certified materials are

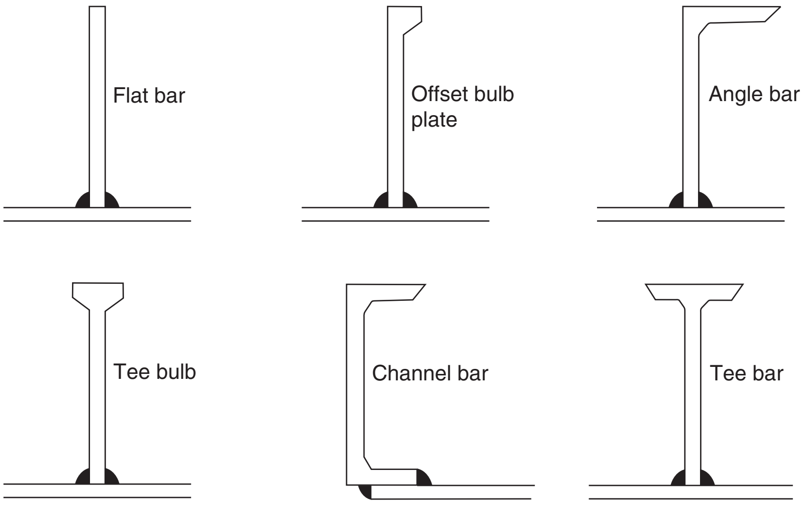


Figure 9.17 Steel sections of shipbuilding.

marked with the Society's brand and other particulars as required by the rules.

Ship classification societies originally had varying specifications for steel: but in 1959, the major societies agreed to standardize their requirements in order to reduce the required grades of steel to a minimum. There are now five different qualities of steel employed in merchant ship construction and now often referred to as IACS steels. These are graded A, B, C, D and E, Grade A being an ordinary mild steel to Lloyds Register requirements and generally used in shipbuilding. Grade B is a better quality mild steel than Grade A and specified where thicker plates are required in the more critical regions, Grades C, D and E possess increasing notch-tough characteristics, Grade C being to American Bureau of Shipping requirements. Lloyds Register requirements for Grades A, B, D and E steels may be found in Chapter 3 of Lloyds Rules for the Manufacture, Testing and Certification of Materials, Lloyds Register (2004).

9.3.2.5 *High tensile steels*

Steels having a higher strength than that of mild steel are employed in the more highly stressed regions of large tankers, container ships and bulk carriers. Use of higher strength steels allows reductions in thickness of deck, bottom shell, and framing where fitted in the midships portion of larger vessels; it does, however, lead to larger deflections. The weldability of higher tensile steels is an important consideration in their application in ship structures and the question of reduced fatigue life with these steels has been suggested. Also, the effects of corrosion with lesser thicknesses of plate and section may require more vigilant inspection.

Higher tensile steels used for hull construction purposes are manufactured and tested in accordance with Lloyds Register requirements. Full specifications of the methods of manufacture, chemical composition, heat treatment, and mechanical properties required for the higher tensile steels are given in Chapter 3 of Lloyds Rules for the Manufacture, Testing and Certification of Materials. The higher strength steels are available in three strength levels, 32, 36, and 40 (kg/mm^2) when supplied in the as rolled or normalized condition. Provision is also made for material with six higher strength levels, 42, 46, 50, 55, 62 and 69 (kg/mm^2) when supplied in the quenched and tempered condition. Each strength level is subdivided into four grades, AH, DH, EH and FH depending on the required level of notch-toughness.

9.3.2.6 *Corrosion resistant steels*

Steels with alloying elements, that give them good corrosion resistance and colloquially referred to as stainless steels are not commonly used in ship structures, primarily because of their higher initial and fabrication costs. Only in the fabrication of cargo tanks containing highly corrosive cargoes might such steels be found.

For oil tankers the inner surfaces, particularly the deckhead and bottom, are generally protected by high cost corrosion resistant coatings that require vigilant inspection and maintenance. A recent development in the manufacture of an alloyed shipbuilding steel with claimed improved corrosion resistance properties and its approval by Lloyds Register for use in certain cargo tanks of a 105 000 dwt tanker indicate that in the future the need to coat oil cargo tanks might be dispensed with.

9.3.2.7 *Steel sandwich panels*

As an alternative to conventional shipyard fabricated stiffened steel plate structures, proprietary manufactured steel sandwich panels have become available and used on ships where their lighter weight was important. Such panels consist of a steel core in the form of a honeycomb with flanges to which the external steel sheets are resistance (spot) or laser (stake) welded. Early use of these bought in steel sandwich panels was primarily for non-hull structures in naval construction where their light weight was important. Also when fabricated using stainless steel their corrosion-resistance and low maintenance properties have been utilized.

A proprietary steel sandwich plate system (SPS) has been developed which consists of an elastomer core between steel face plates. Elastomers are a specific class of polyurethane that has a high tolerance to mechanical stress i.e. it rapidly recovers from deformation. The SPS elastomer also has a high resistance to most common chemical species. Initial application of SPS in shipbuilding has been in passenger ship superstructures where the absence of stiffening has increased the space available and provided factory finished surfaces with built in vibration damping, acoustic insulation and fire protection. SPS structures have been approved with an A 60 fire-resistance rating. Also SPS overlays have been applied to repair existing work deck areas. SPS structures can be fabricated using joining technologies presently used in the shipbuilding industry, but the design of all joints must take into account the structural and material characteristics of the metal-elastomer composite. The manufacturer envisages the use of SPS panels throughout the hull and superstructure of ships providing a simpler construction with greater carrying capacity and less corrosion, maintenance and inspection. In association with the manufacturer Lloyds Register in early 2006 published provisional Rules for the use of this sandwich plate system for new construction and ship repair. The Rules cover construction procedures, scantling determination for primary supporting structures, framing arrangements and methods of scantling determination for steel sandwich panels.

The Norwegian classification society, Det Norske Veritas (DNV), have proposed for bulk carrier hulls the use of a lightweight concrete/steel sandwich. They envisage a steel/concrete/steel composite structure for the cargo hold area of say 600 mm width for the side shell but somewhat greater width for the double bottom area. This sandwich would be much narrower than for a comparable steel-only double skin bulk carrier thus increasing the potential carrying capacity although water ballast may have to be carried in some designated holds as the double skin would not be available for this purpose. DNV consider the other advantages of the concrete/steel sandwich to be reduced stress concentrations with less cracking in critical areas, considerable elimination of corrosion and elimination of local buckling. At the time of writing DNV were undertaking a two-year investigation programme in association with a shipyard to study the practicalities of their sandwich proposal.

9.3.2.8 *Steel castings*

Molten steel produced by the open hearth, electric furnace, or oxygen process is pored into a carefully constructed mould and allowed to solidify to the shape required. After removal from the mould a heat treatment is required, for example annealing, or normalizing and tempering to reduce brittleness. Stern frames, rudder frames, spectacle frames for bossings, and other structural components may be produced as castings.

9.3.2.9 *Steel forgings*

Forging is simply a method of shaping a metal by heating it to a temperature where it becomes more or less plastic and then hammering or squeezing it to the required form. Forgings are manufactured from killed steel made by the open hearth, electric furnace, or oxygen process, the steel being in the form of ingots cast in chill moulds. Adequate top and bottom discards are made to ensure no harmful segregations in the finished forgings and the sound ingot is gradually and uniformly hot worked. Where possible the working of the metal is such that metal flow is in the most favourable direction with regard to the mode of stressing in service. Subsequent heat treatment is required, preferably annealing or normalizing and tempering to remove effects of working and non-uniform cooling.

9.3.3 Aluminium alloy

9.3.3.1 *General*

There are three advantages which aluminium alloys have over mild steel in the construction of ships. Firstly aluminium is lighter than mild steel (approximate weight being aluminium 2.723 tonnes/m^3, mild steel 7.84 tonnes/m^3), and with an aluminium structure it has been suggested that up to 60% of the weight of a steel structure may be saved. This is in fact the principal advantage as far as merchant ships are concerned, the other two advantages of aluminium being a high resistance to corrosion and its non-magnetic properties. The non-magnetic properties can have advantages in warships and locally in way of the magnetic compass, but they are generally of little importance in merchant vessels. Good corrosion properties can be utilized, but correct maintenance procedures and careful insulation from the adjoining steel structure are necessary. A major disadvantage of the use of aluminium alloys is their higher initial and fabrication costs. The higher costs must be offset by an increased earning capacity of the vessel, resulting from a reduced lightship weight or increased passenger accommodation on the same ship dimensions. Experience with large passenger liners on the North Atlantic service has indicated that maintenance costs of aluminium alloy structures can be higher for this type of ship and service.

A significant number of larger ships have been fitted with superstructures of aluminium alloy and, apart from the resulting reduction in displacement, benefits have been obtained in improving the transverse stability. Since the reduced weight of superstructure is at a position above the ship's centre of gravity this ensures a lower centre of gravity than that obtained with a comparable steel structure. For example on the Queen Elizabeth 2, with a limited beam to transit the Panama Canal, the top five decks constructed of aluminium alloy enabled the ship to support one more deck than would have been possible with an all steel construction.

Only in those vessels having a fairly high speed and hence power, also ships where the deadweight/lightweight ratio is low, are appreciable savings to be expected. Such ships are moderate – and high – speed passenger liners having a low deadweight. It is interesting to note however that for the Queen Mary 2, not having a beam limitation, the owners decided to avoid aluminium alloy as far as possible to ensure ease of maintenance over a life cycle of 40 years. A very small number of cargo liners have been fitted with an aluminium alloy superstructure, principally to clear a fixed draught over a river bar with maximum cargo.

The total construction in aluminium alloy of a large ship is not considered an economic proposition and it is only in the construction of smaller multihull and other high speed craft where aluminium alloys higher strength to weight ratio are fully used to good advantage.

9.3.3.2 *Production of aluminium*

For aluminium production at the present time the ore, bauxite, is mined containing roughly 56% aluminium. The actual extraction of the aluminium from the ore is a complicated and costly process involving two distinct stages. Firstly the bauxite is purified to obtain pure aluminium oxide known as alumina; the alumina is then reduced to a metallic aluminium. The metal is cast in pig or ingot forms and alloys are added where required before the metal is cast into billets or slabs for subsequent rolling, extrusion, or other forming operations.

Sectional material is mostly produced by the extrusion process. This involves forcing a billet of the hot material through a die of the desired shape. More intricate shapes are produced by this method than are possible with steel where the sections are rolled. However, the range of thickness of section may be limited since each thickness requires a different die. Typical sections are shown in Figure 9.18.

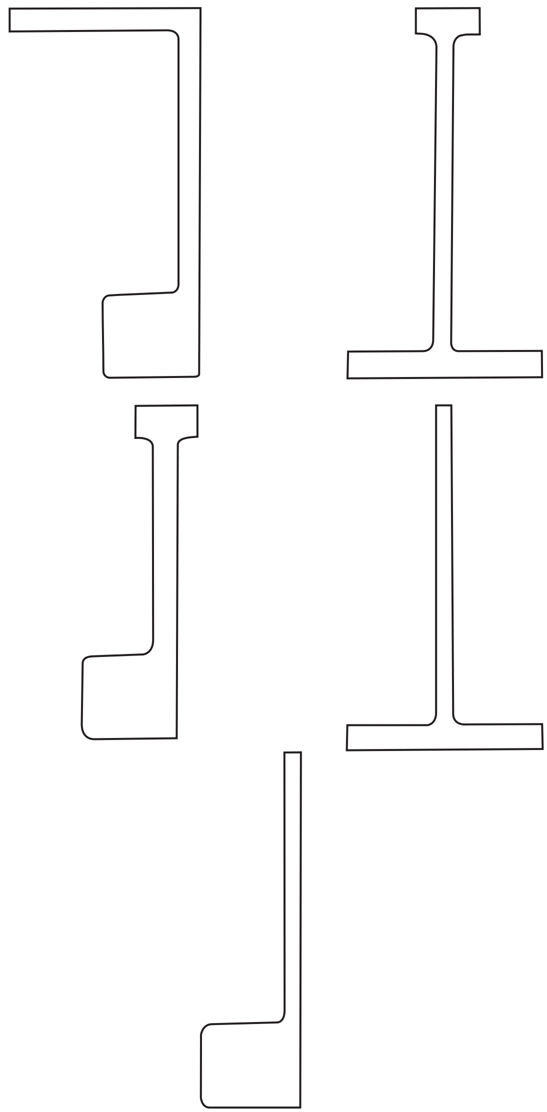


Figure 9.18 Typical aluminium alloy sections.

Table 9.4 Alloying elements.

Element	*5083*	*5086*	*6061*	*6082*
Copper	0.10 max	0.10 max	0.15–0.40	0.10 max
Magnesium	4.0–4.9	3.5–4.5	0.8–1.2	0.6–1.2
Silicon	0.40 max	0.40 max	0.4–0.8	0.7–1.3
Iron	0.40 max	0.50 max	0.70 max	0.50 max
Manganese	0.4–1.0	0.2–0.7	0.15 max	0.4–1.0
Zinc	0.25 max	0.25 max	0.25 max	0.20 max
Chromium	0.05–0.25	0.05–0.25	0.04–0.35	0.25 max
Titanium	0.15 max	0.15 max	0.15 max	0.10 max
Other elements				
each	0.05 max	0.05 max	0.05 max	0.05 max
total	0.15 max	0.15 max	0.15 max	0.15 max

Aluminium alloys. Pure aluminium has a low tensile strength and is of little use for structural purposes; therefore the pure metal is alloyed with small percentages of other materials to give greater tensile strengths, Table 9.4. There are a number of aluminium alloys in use, but these may be separated into two distinct groups, non-heat treated alloys and heat treated alloys. The latter as implied are subjected to a carefully controlled heating and cooling cycle in order to improve the tensile strength.

Cold working of the non-heat treated plate has the effect of strengthening the material and this can be employed to advantage. However, at the same time the plate becomes less ductile, and if cold working is considerable the material may crack; this places a limit on the amount of cold forming possible in shipbuilding. Cold worked alloys may be subsequently subjected to a slow heating and cooling annealing or stabilizing process to improve their ductility.

With aluminium alloys a suitable heat treatment is necessary to obtain a high tensile strength. A heat treated aluminium alloy which is suitable for shipbuilding purposes is one having as its main alloying constituents magnesium and silicon. These form a compound Mg_2Si and the resulting alloy has very good resistance to corrosion and a higher ultimate tensile strength than that of the non-heat treated alloys. Since the material is heat treated to achieve this increased strength, subsequent heating, for example welding or hot forming, may destroy the improved properties locally, Kecsmar and Shenoi (2004).

Aluminium alloys are generally identified by their Aluminium Association numeric designation. The 5000 alloys being non-heat treated and the 6000 alloys being heat treated. The nature of any treatment is indicated by additional lettering and numbering. Lloyds Register prescribe the following commonly used alloys in shipbuilding:

5083-0	annealed
5083-F	as fabricated
5083-H321	strain hardened and stabilized
5086-0	annealed
5086-F	as fabricated
5086-H321	strain hardened and stabilized
6061-T6	solution heat treated and artificially aged
6082-T6	solution heat treated and artificially aged

Riveting. Riveting may be used to attach stiffening members to light aluminium alloy plated structures where appearance is important and distortion from the heat input of welding is to be avoided.

The commonest stock for forging rivets for shipbuilding purposes is a non-heat treatable alloy NR5 (R for rivet material) which contains 3–4% magnesium. Non-heat treated alloy rivets may be driven cold or hot. In driving the rivets cold relatively few heavy blows are applied and the rivet is quickly closed to avoid too much cold work, i.e. becoming work hardened so that it cannot be driven home. Where rivets are driven hot the temperature must be carefully controlled to avoid metallurgical damage. The shear strength of hot driven rivets is slightly less than that of cold driven rivets.

9.3.3.3 *Aluminium alloy sandwich panels*

As with steel construction, proprietary aluminium alloy honeycomb sandwich panels are now available to replace fabricated plate and stiffener structures and can offer extremely low weight options for the superstructures of high speed craft.

9.3.3.4 *Fire protection*

It is considered necessary to mention when discussing aluminium alloys that fire protection is more critical in ships in which this material is used because of the low melting point of aluminium alloys. During a fire the temperatures reached may be sufficient to cause a collapse of the structure unless protection is provided. The insulation on the main bulkheads in passenger ships will have to be sufficient to make the aluminium bulkhead equivalent to a steel bulkhead for fire purposes.

For the same reason it is general practice to fit steel machinery casings through an aluminium superstructure on cargo ships.

9.3.4 Composite materials

9.3.4.1 *Overview*

In this section a summary is made of the design of marine structures made from composite materials. Attention is focused on fibre-reinforced plastics (FRP), but it should be noted that the term 'composites' can include materials such as fibre-reinforced metals, fibre-reinforced cement and combinations of FRP, wood, metal and concrete see also Section 9.3.2.7. This section on the marine applications of FRP composite materials has been taken from Shenoi and Dodkins (2000). Further developments in the properties and applications of composites can be found in references such as Shenoi and Wellicome (1993), Clarke *et al.* (1998), Jeong and Shenoi (2001), Kelly and Zweben (2000), Backman (2005) and Vasiliev and Morozov (2007).

9.3.4.2 *Introduction*

Polymeric composite materials have been used in ships, boats, and other marine structures for over 50 years, Smith (1990), Shenoi and Wellicome (1993). The motivation for their use has varied from application to application. In naval minehunters, for instance, the main driver for their usage is the non-magnetic and non-conducting capability of glass reinforced plastics (GRP). In the case of dinghies, canoes, and small harbour craft, GRP is preferred because of competitive first cost and the ease with which complex shapes required for such craft can be fabricated. Yet another factor leading to increased use is the good fire resistance of fibre reinforced plastics (FRP) – this is so with regard to applications in offshore structures. Other issues encouraging the increased use of FRP are: (i) low operating (maintenance) cost; (ii) good fatigue resistance; (iii) high specific strength; (iv) good corrosion resistance; (v) good thermal resistance; and (vi) reduced parts count.

The purpose of this section is to provide a broad overview to the design of ship structures made from composite materials. As a precursor to this, it is essential to understand certain key features distinguishing ships and marine structures. Ships are products of a one-off variety. There are very few series of similar ships. The largest production run could be of the order of about 10–12 ships of one kind. This is a very rare occurrence, though. What this means is that the design effort has to be dedicated for each ship order. This contrasts with a large-volume production of an aircraft type (e.g., Boeing 747, Airbus A320, etc.) or a motor car model (e.g., Ford Fiesta, Volvo 460 series, etc.), where the design effort can be focused to a greater degree.

The lead time for ships, from order to delivery, is very short. For large tankers and bulk carriers this can be as short as a few months. For naval ships, such as minehunters, this can be about three to four years. For smaller craft, the time span can vary from a few months to a couple of years. This places a tremendous pressure on marine designers and production engineers to produce practical and cost-effective solutions rapidly.

Ships and other marine craft, with a few exceptions, are generally low-cost modes of transport. The cargo freight rates or passenger ticket prices for the marine mode are several orders of magnitude smaller than for aircraft. This implies that ships have to have a much lower life cycle cost. A significant component of this economic balance is primary (or production) cost. Thus marine designers need to search for solutions using relatively inexpensive materials in production processes which do not require substantial tooling or other forms of high-cost infrastructural investment.

The implication of the above-listed three features is that marine design has to be done rapidly, using technology that is well proven and with relatively large factors of safety to account for uncertainties in a variety of production and operational areas. This section seeks to provide an overview of marine structural design in composite materials, particularly with regard to materials selection, design procedures, structural synthesis, and external influences on design.

9.3.4.3 *Materials selection*

(a) Materials selection

1. Reinforcements

For marine applications, generally the choice of reinforcement is simplified because cost constraints render the more expensive high-performance reinforcements such as carbon and aramid unattractive. The emphasis for bulk use is strongly on glass fibre. This has been used in a variety of forms including unidirectional tows, woven and stitched fabrics, and chopped random mats. There are some areas in high-performance craft where combinations of carbon and aramid fibres are being considered, Serter (1997), Maccari and Farolfi (1992). However, glass still accounts for over 95% of the usage in marine applications.

Some key property parameters influencing the selection of structural materials for marine use, Gibson (1993), are shown in Figures 9.19 and 9.20. Figure 9.19 compares various materials in terms of strength per unit weight and stiffness per unit weight. It is evident that composites have better characteristics than metals with regard to specific strength. However, in terms of specific stiffness, only carbon and aramid composites outperform metals. Glass-based composites are more flexible. Apart from mechanical performance, structural materials also have cost implications. In Figure 9.20 it is clear that none of the composites is competitive with metals in stiffness-critical areas. Furthermore, in strength-critical cases, only glass-based composites can compete with metals. This is the underlying reason for the large usage of glass in large-volume applications such as ships, offshore structures, and other marine artifacts.

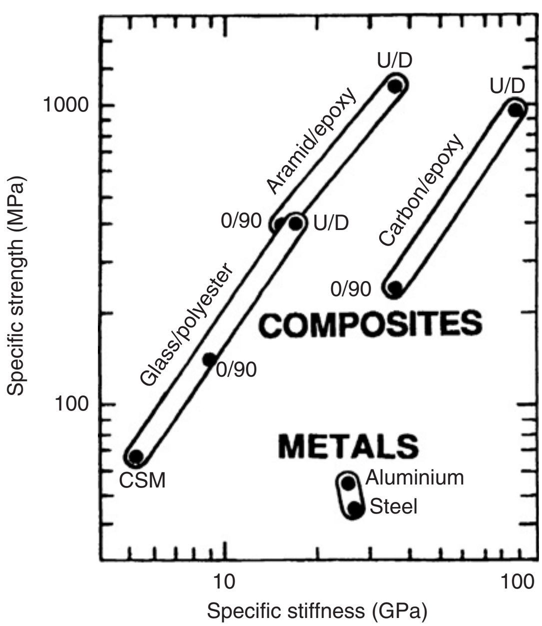


Figure 9.19 Specific strength and stiffness properties for typical engineering materials, Gibson (1993).

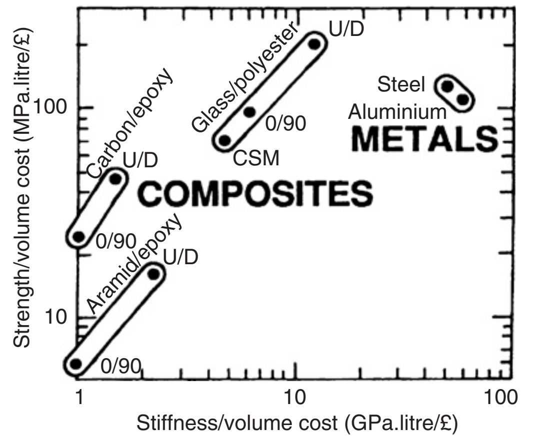


Figure 9.20 Strength and stiffness per volume cost for typical engineering materials, Gibson (1993).

The mechanical properties are engineered by the appropriate use of different forms of the reinforcement, (see Figure 9.21). High fibre volume fractions are desirable in some applications and certain regions of ships. Unidirectional rovings give the highest fibre volume fraction, usually in the range 0.5–0.65. In woven fabrics, the volume fraction is generally 0.4–0.55, while with random mats only 0.25–0.33 is achievable.

2. Matrix resins

The matrix plays a critical role in determining off-axis strength, damage tolerance, corrosion resistance, and thermal stability. Current technology and cost constraints limit the selection to thermosets and there are three widely used candidates, as shown in Table 9.5, each with particular strengths and drawbacks.

Unsaturated polyesters are the most widely used resins in the marine industry. Their principal advantage, apart from low cost, lies in their cure chemistry. The free radical cure reaction, triggered by the addition of a peroxide catalyst, offers a rapid but controllable cure, while the resins themselves have a long shelf life. For this reason, polyesters are easily fabricated. Among the various types of polyester resins, the isophthalic variety offers the most attractive combination of mechanical strength and resistance to the marine environment. However, from a cost viewpoint, the orthophthalic variety holds attractions for the small boat industry.

Vinyl ester resins lie midway between polyesters and epoxies. While retaining some of the fabricability of the free radical cure, they offer better mechanical properties and are often preferred in demanding

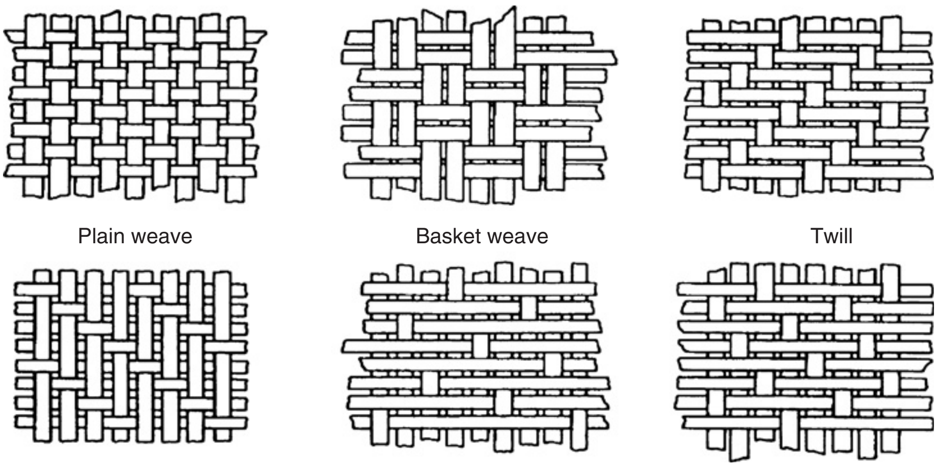


Figure 9.21 Types of fibre reinforcement with potential use in marine applications.

Table 9.5 Candidate resins for use in marine applications.

Resin	*Cost* (£/tonne)	*Mechanical strength*	*Corrosion resistance*	*Fire performance*
Polyester	1200–1600	xx	xx	x
Vinyl ester	2200–2600	xxx	xxx	x
Epoxy	>4000	xxxxx	xxxxx	x

applications, particularly where chemical or environmental resistance is needed.

Epoxy resins, of which there are several variants, offer the most outstanding combination of strength, toughness, and corrosion resistance. They are, however, expensive. Fabrication can also be more difficult and hazardous compared with polyesters and vinyl esters. They are most widely used with higher performance fibres in vessels where high strength, toughness, and damage tolerance are prime requirements.

3. Core materials

The choice in this context is mainly between PVC foams, balsa wood, and honeycomb materials. Expanded closed-cell polyvinyl chloride (PVC) foam has been widely used in many marine applications. It is available in a range of densities, varying from 45 to over 200 kg m^{-3}. There are also several varieties of these, including linear PVC which has high ductility but low mechanical properties and cross-linked PVC which has high strength and stiffness but is relatively brittle. PVC foams offer good resistance to water penetration, good thermal and electrical insulation, and effective vibration and damping characteristics. Their main deficiencies are reduction of strength and stiffness at modestly elevated temperatures (typically a loss of 50% of compressive and shear moduli and strengths at temperatures in the range 40–60°C), outgassing at temperatures up to 100°C and chemical breakdown, with emission of HCl vapour at temperatures of over 200°C.

End grain balsa is one of the most efficient, Hearmon (1948), and moderately priced sandwich core materials. Its main deficiency is susceptibility to water penetration and consequential swelling, debonding, and rot. Although some success has been claimed for the balsa core sandwich construction in boats, Lippay and Levine (1968), a number of disastrous instances of water penetration and subsequent deterioration of balsa core have also occurred. For these reasons, use of balsa core in the primary hull and deck structure of ships and boats is not normally advisable.

Sandwich panels and shells with ultralight honeycomb cores in aluminium, FRP, or resin-impregnated paper, developed in many cases for aerospace structures, are generally too expensive for marine construction. However, they have a limited application in decks and bulkheads of weight-critical craft such as hydrofoils, hovercraft, and high-performance sailing yachts and in specialized components such as radomes. They are unlikely to be suitable for the primary hull structure of high-speed craft because of the risk of water penetration and core-skin debonding under impact loads, though they have recently found extensive application in racing yachts where robustness and durability are sacrificed in favour of performance.

With all forms of sandwich construction, regardless of core materials used, a sound and consistent bond between core and the skins is of paramount importance. Production techniques, quality control and inspection methods need to be applied with great care on ship scale sandwich fabrication.

(b) Mechanical properties

1. Static properties

These may be carried out using BS, ISO, European, ASTM or naval standards, Sims (1993). A selection of these are listed below.

Component fractions	BS2782, ISO 1172/7822, ASTM 2584/2374/ D3171
Tensile	BS2782, ISO3268, EN2597, ASTM 2585
Compressive	ISO 8515/604, ASTM D3410, EN 2850
Flexural	ISO 178, ASTM D790, BS 2782, EN 2561
In-plane shear	ASTM D3518/4255/3846
Interlaminar shear	ISO 4585, BS 2782, EN2563, ASTM D2344
Sandwich materials	ASTM C273/297/364/365/ 393/394/408

Laminates incorporating woven reinforcement are tested in both the warp and weft directions. It is important to emphasize that variability in properties may be high especially for marine laminates. The values of strength and modulus to be used in design calculations should correspond to ‘mean minus two standard deviations’ limit derived from mechanical test data. This implies a 97.5% probability that the design figure will be exceeded in the actual structure.

2. Long-term properties

All resins absorb a certain small percentage of moisture. The general pattern of degradation in a marine laminate is an initial fall of 10–20% in mechanical properties in the first 12–18 months followed by a very slow fall over the rest of the period. However, if a saturated laminate is subjected to continuous tensile stress, then the degradation may be more severe. Laboratory experiments have shown that sustained stress levels should not exceed about 20% of the ultimate values, Dodkins (1993). This is generally not a problem in ships’ structures designed for extreme load cases due to wave action etc., but special attention should be paid to structure supporting dead loads. It is advisable to keep strain levels below those at which resin microcracking occurs, say 0.3–0.5%, thus avoiding moisture ingress.

Accelerated aging tests are also now coming into more prominence. The acceleration is usually achieved by increasing the temperature above ambient, though well below the heat distortion temperature. This is typically 60–70°C for a non-post cured polyester laminate. Quicker results may be obtained by heating to 80–90°C and comparing the performance to that of a laminate with known resistance to aging.

In specifying the lay-up of a shell laminate of the ship’s hull, an all-woven roving configuration is acceptable, though the outer surface must contain a layer of chopped strand mat. The mat takes up a greater proportion of resin than the roving and short fibres ensure that if the glass becomes exposed on the surfaces, water cannot wick far along the fibres thus ensuring that any minor surface damage as a result of impact and abrasion remains localized. The mat layer therefore forms a protective barrier to the underlying woven layers. The use of gel coats on yachts perform the same function, while giving a good aesthetic finish at the same time.

3. Fire resistance

The fire resistance of a structure is difficult to characterize accurately, the requirements being generally: (i) maintenance of strength and stiffness until a fire is extinguished; (ii) limitation of temperature and prevention of spread of flames to adjacent compartments; and (iii) minimization of smoke and toxic fumes.

As with all polymeric materials, FRP is combustible. However, FRP, although flash-igniting in air temperatures of 350–400°C, burns slowly, is readily extinguished by water sprinkling or oxygen exclusion, and providing woven reinforcement is used, provides a partially effective barrier in the form of exposed glass fibres. Because of its low conductivity,

Table 9.6 Fire-related properties of metals and FRP.

Material	*Melting temperature* (°C)	*Thermal conductivity* w/(m.°C)	*Heat distortion temperature* (°C) (BS2752)	*Self-ignition temperature* (°C)	*Flash ignition temperature* (°C)	*Oxygen index* (%) (ASTM D2863)	*Smoke density* D_m (ASTM E662)
Aluminium	660	240	–	–	–	–	–
Steel	1430	50	–	–	–	–	–
E-Glass	840	1.0	–	–	–	–	–
Polyester resin	–	0.2	70	–	–	20–30	–
Phenolic resin	–	0.2	120	–	–	35–60	–
GRP (polyester based)	–	0.4	120	480	370	25–35	750
GRP (phenolic based)	–	0.4	200	570	530	45–80	75

GRP meets requirement (ii) effectively. Emission of smoke and fumes by burning polyester resin presents a serious problem which requires special ventilation and fire-fighting facilities. An increased effectiveness can be provided by the use of intumescent and other fire-retardant coatings which can be incorporated in the surface of the laminate as a gel coat.

Phenolic-based FRP, which offers a high level of strength and stiffness retention at temperatures of up to 250°C, high flash-ignition temperature (about 530°C), and oxygen index (>45%), together with low smoke and toxic fume emission, should be considered carefully for fire-critical structures such as bulkheads in accommodation areas and machinery compartments.

Some of the fire-related characteristics of marine structural materials are listed in Table 9.6, Dow and Bird (1994), Don and Bird (1994). Fire is an important issue which is currently affecting the increased use of composites in marine vessels; legislative issues covering this aspect are discussed in more detail in Section 9.3.4.6.

(c) Production considerations

1. Production processes

(i) Open mould wet lay-up (hand lay-up)

Until very recently, production of marine structures was achieved using one of two techniques. Small, 'low-tech' dinghies, yachts, and similar craft have been produced by the spray-gun technique, where reinforcement strands and resin are injected together on to the surface of a mould. This results in a randomly oriented reinforcement in a resin-rich form. Larger ships have been built using the hand lay-up techniques. Some automation in resin impregnation was achieved even in the early days of GRP shipbuilding, Smith (1990), though this was limited to some shipyards and certain parts of the structure.

(ii) Vacuum-assisted resin infusion moulding

The most significant breakthrough in FRP fabrication in a marine context occurred in the early 1990s with the introduction of vacuum-assisted resin infusion moulding. This is now gradually replacing the wet lay-up process in a comprehensive manner. Vosper Thornycroft (now UT), for instance, used SCRIMP (or Seeman Composites Resin Infusion Moulding Process) for the production of large ship scale mouldings. Bulkheads and plate panels up to 10 m × 10 m in size and 18 mm thickness are currently in production. Hull and superstructure mouldings up to 30 m long have also been successfully produced.

Such resin infusion techniques have distinct advantages over the wet lay-up process.

(i) High compaction under vacuum results in laminates of high quality and fibre content and enhanced mechanical properties with improved uniformity.
(ii) Air voids are virtually eliminated.
(iii) A cleaner production process is achieved with very low styrene emissions.
(iv) Electromagnetic screening in the form of a metallic mesh can be embedded in the lay-up prior to resin infusion so that it can become an integral part of the structure.
(v) A weight saving of 15% has been achieved on single skin parts.
(vi) For sandwich construction, both skins can be wetted-out and bonded to the core in a single infusion process.

2. Practical considerations

(i) Wet lay-up

FRP laminating should take place inside climate controlled buildings. Polyester resin cures at room temperature (above 16°C). A styrene fume extraction system maintains shop airborne styrene levels in compliance with regulatory requirements. This covers the requirements imposed by the wet lay-up process, where significant quantities of styrene are released through evaporation from large areas of exposed wet laminate. It is noteworthy that in the vacuum-assisted resin infusion process, the laminate is sealed under a nylon film and nearly all styrene is cross-linked during the curing process.

The characteristics of laminating materials affect both the quality of the final laminate and the production time. These features may be evaluated realistically only by large-scale production trials (which are discussed in the next section). Important aspects to consider in materials selection are discussed below.

(i) Ease of cutting the reinforcement cloth is largely dependent on the cloth weight. If it is intended to use a range of standard widths, then it is advantageous to have the material supplied ready cut to those widths with the edges stitched in order to prevent fraying. Woven roving is more likely to fray than combination cloth where the stitching and mat layer hold the cut edges.

(ii) While most cloths will wrap easily around a cylindrically shaped mould surface, not all will form around a corner or a shape with double curvature without some tailoring. Examples are the snapped ends of stiffeners, tapered stiffener sections, and the bow section of the hull. Cloths with poor drapeability should be avoided as they lead to excessive tailoring which results in cloth joints that are too close together, necessitating additional material to compensate for the loss in strength. Conversely, a cloth that is very drapeable is too easily distorted such that the rovings are pulled out of a straight line. Coloured threads may be incorporated with the warp rovings to help maintain the straightness during lay-up.

(iii) The use of a heavy cloth implying the need to have a reduced number of plies does not always lead to reduced production time. Each of the heavier cloth layers will take longer to set up and consolidate. In any case, there is a limit on the weight of cloth and resin that can be laid wet-on-wet at one time (see the issue below on resin curing).

(iv) A low viscosity resin reduces the time taken for consolidation (wet-out of the cloth and removal of air bubbles by rolling). The resin should be thixotropic to reduce drainage on vertical surfaces. However, this tends to conflict with the ease of wetting out. This feature is particularly important in resin selection.

(v) The curing reaction of the polyester resin is exothermic. In the wet lay-up process, if the laminate thickness build-up is too fast, then the later layers tend to insulate the earlier layers and prevent dissipation of the heat. The laminate then begins to heat up more, which further accelerates the curing reaction until a runaway situation develops and the temperature may rise to a point where the laminate becomes permanently heat damaged.

Before finalization of a lay-up for any structure, it is desirable to carry out production trials. Initial trials may be carried out on small panels measuring about $1\,\text{m} \times 2\,\text{m}$. These should have any envisaged stiffening to be bonded to the surface to check drapeability over the sides and ends of the stiffening. Handling may be evaluated to a limited extent and the resin ratio and ply thicknesses can also be checked. At this early stage in the materials selection process, a large number of materials can be evaluated economically.

Having short-listed the materials with adequate handling characteristics, panels of about $3\,\text{m} \times 3\,\text{m}$ could be fabricated; these samples should be used to cut samples for testing to determine mechanical properties. Panels of this size are important because they reflect the level of difficulty involved in large-scale production; laminates will normally include a realistic void content and butted and staggered cloth edges.

Once the mechanical tests have been completed, the number of fibre/resin combinations can be reduced to perhaps two or three. These materials may now be tested again in more realistic production trials. Short sections of the hull, perhaps three to four frame spaces long as a minimum, should be laid up in the hull mould. These sections should include examples of all principal structural features of the proposed design such as frames, bulkheads, stiffeners, tee joints, beam knee joints, stiffener-to-shell connections, etc.

(ii) Vacuum-assisted resin infusion moulding

Mould surfaces used for this process need to be completely airtight as full vacuum is applied to the entire moulding area. However, unlike conventional resin transfer moulding, no positive resin injection pressure is applied, so that moulds need not be heavily reinforced. A drawback though is the higher cost of consumable materials such as tubing and flow medium and resin lost in feed and vacuum tubes. However, reusable bags have been developed to suit series production of small to medium sized mouldings.

Good control of the workshop environment, resin, and mould temperatures must be maintained for SCRIMP to produce consistent results. The process very much depends on knowing gel times, which are largely temperature dependent.

When producing large mouldings, thought must be given to the means of catalyzing and infusing large quantities (perhaps over 1 tonne) of resin in timescales of 40min or less. This requires a high level of shop floor planning and teamwork.

Many of the comments made earlier with regard to selection of constituent materials also apply to SCRIMP. However, compared with wet lay-up, low viscosity resins are preferred in order to maximize flow rate and as all plies of fabric representing the full lay-up are applied dry and infused together, the weight of individual fabric layers is not important, thereby allowing the specification of fewer plies of heavier fabrics in order to reduce the manhours required for mould preparation.

For any new lay-up or materials, production trials are required to finalize the infusion set-up and procedure, measure flow rates across the mould, and select the gel time required. This is best carried out on a glass-topped moulding table such that the resin flow front can be viewed and timed from above and below the laminate. The results of such trials are normally sufficient to scale up to full-sized production mouldings, provided the effects of all features of the production mouldings have been checked.

9.3.4.4 *Design concepts*

(a) Design spiral

Ship design and, by inference, ship structural design, is iterative in nature (Figure 9.22). This is specifically derived for preliminary design purposes when the ship framing on the decks and bottom is longitudinal in nature (see Section 9.3.4.5(c)); it typifies one aspect of the more all-inclusive process of overall ship design and is therefore a spiral within spirals. Inferences to be drawn from the illustration are that among the interlocking constraints which must be satisfied, albeit in harmony with each other, the web frame and longitudinal spacings are tentatively set as initial conditions on which the final, optimized design is to be based. Other optimized designs for varying frame spacings could also be investigated.

First estimates of plate thicknesses, section details, joint specifications, longitudinal scantlings, and materials choices will by necessity be rough. However, as the process proceeds towards convergence, more characteristics of the design become known and, hence, more refined methods can be used. The progression from approximate analytical expressions to more refined finite element analysis based techniques is discussed in more detail later in this section. With each iterative cycle, necessary modifications from the one previous become smaller. A designer has to make the decision to finalize the design, after a requisite number of iterations, having met all performance-based requirements, based on its ability to be produced in a cost-efficient manner and to be maintainable at reasonable costs during the operational life of the ship.

(b) Design loads

1. General

The first step in any structural design is to define the loads that will act on the structures. For ship and boat design, this exercise can be exhaustive and tedious. Primary loads from the operation of the vessel in the seaway must ideally allow for the variability of the ocean environment itself. Secondary and tertiary loads resulting from locally-induced sources such as the main engines, heavy cargo in one compartment, etc. may also be critical in some cases. In addition to the magnitude and direction of the loads, it is also important to know the frequency with which the force systems acts in order that fatigue calculations may be adequately carried out. Details of such calculations may be learnt from Chapter 4 or any standard naval architectural textbook, Lewis (1988). Table 9.7 lists the principal loads to be considered in warship design, for instance.

There are a variety of guides available from regulatory authorities in various countries such as Lloyd's Register of Shipping (1998a), American Bureau of Shipping (1998). These are increasingly based on first-principles mechanics concepts, though because of the very nature of uncertainties associated with seaways, there is still a significant reliance on empiricism and operational experience.

The purpose of this section is to outline the principal types of loads and their characteristics. Loads to be considered here include:

(i) hull girder bending loads that act over the entire length of the ship;
(ii) wave slamming loads on ships and high-speed craft;
(iii) deck and bulkhead loads;
(iv) point loads.

These aspects are described in more detail in Chapter 4.

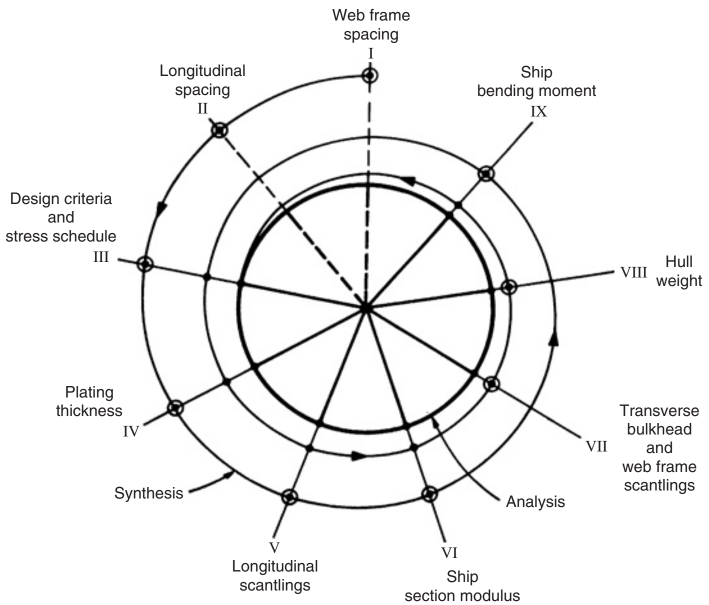


Figure 9.22 Structural design spiral.

Table 9.7 Loads imposed on warship structures.

Basic loads	*Sea loads*	*Operational loads*	*Combat loads*
Live loads	Hull bending	Flooding	Primary Shockwave
Structure self-weight	Wave slamming	Helicopter landing	Gun blast pressures
Tank pressures	Roll/pitch/heave inertia	Replenishment at sea	Explosion-induced whipping
Equipment weights	Wind loads	Docking	Fragmentation
		Anchoring	Gun recoil
		Berthing	Missile efflux pressures

2. The hull as a longitudinal girder

Classical approaches to ship structural design treat the hull structure as a beam for purposes of analysis. The validity of this approach is related to the vessel's length-to-beam (L/B) and length-to-depth (L/D) ratios. Hull girder methods are applied to L/D values greater than 12. From practical considerations this refers to vessels greater than about 50 m in length.

(i) Still water bending moment

Before a ship even goes out to sea, some stress distribution profile exists within the structure. Figure 9.23 shows how the summation of buoyancy and weight distribution curves of an idealized rectangular barge lead to shear force and bending moment distribution diagrams. Stresses apparent in the still water condition generally become extreme only in cases where concentrated loads are applied to the structure, which can be the case when the holds of a cargo vessel are selectively filled.

(ii) Wave bending moment

A quasistatic approach to predicting stresses in a seaway involves the superposition of a trochoidal wave with a wavelength equal to ship length in the hogging and sagging conditions (see Figure 9.24). The wave height is usually taken as $L/10$ ($L < 60$ m), $L/15$ ($60\,\text{m} < L < 90\,\text{m}$), $L/20$ ($90\,\text{m} < L < 150\,\text{m}$), and $0.6\ L^{0.6}$ ($L > 150$ m). Except for very slender craft, this will not apply to smaller vessels. See also Section 4.1.2.14.

3. Dynamic forces on large ships

(i) Ship oscillation forces

The dynamic response of a vessel operating in a given sea spectrum is very difficult to predict

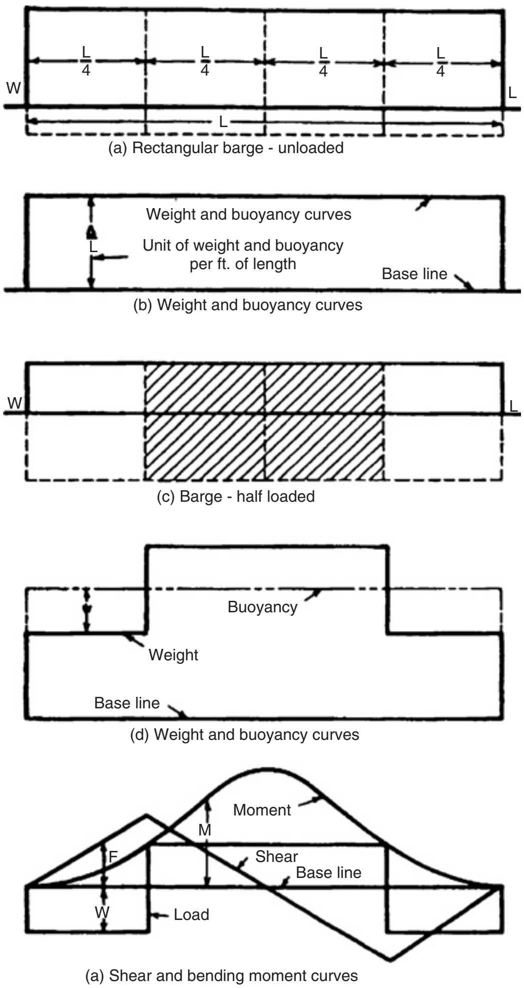


Figure 9.23 Still water bending moment distribution in an idealized rectangular barge.

analytically. Accelerations experienced in the vessel vary as a function of the vertical, longitudinal, and transverse location. These accelerations produce virtual increases of the weight of the concentrated masses, resulting in consequential increases in stress. The designer should have a feel for the worst locations and the type of dynamic behavior that can produce extreme load scenarios. It is generally assumed that combined roll and pitch forces near the deck edge forward represents a (worst case) condition for the extreme accelerations for the ship. There are a number of two- and three-dimensional codes that are used for determining the dynamic response of ship hull girders, Bishop and Price (1979), Faltinsen (1992). In the main, these are used for analytical purposes: for design synthesis, reliance is still placed to a large degree on classification society rules, Lloyd's Register of Shipping (1998a), American Bureau of Shipping (1998).

(ii) Dynamic phenomena

This is principally related to high-frequency loading such as vibrations. Such loading can be either steady state, as with propulsion system induced phenomena, or transient, such as slamming through waves. In the former case, load amplitudes are generally within the design limits of the hull structural material choice. However, repetitive loading implies that fatigue can be a significant issue. Further, a preliminary vibration analysis of major structural elements (such as the hull girder, engine foundations, deck houses, masts, etc.) is generally prudent to ensure that the natural frequencies are not near the propeller shaft or propeller blade rotation rate for normal operating modes. See Section 4.3.

4. Wave slamming on small craft

Slamming is defined in the classical sense as 'high impulsive water pressures at certain speeds in severe seas when the ship motions become large enough to result in forefoot emergence', Ochi and Motter (1973). The timescale during which a slamming pressure acts on a panel is very short, of the order of hundredths of a second. At a given point on a ship panel, however, the maximum pressure will be present only for a few thousandths of a second. This is of particular relevance in the context of composite ships because FRP composite panels have eigenfrequencies of the same order of magnitude as the frequency of the peak-slamming load. Although Ochi and Motter (1973) estimated that more than 300 papers on slamming had already been published by the 1970s, the complexity of the phenomenon is such that universally applicable analytical solutions are still beyond the designers' capabilities.

On a practical level, slam load prediction is still done on the basis of the pioneering work by Heller and Jasper (1960). The method is based on relating strain in a structure from a static load to the corresponding value from dynamic conditions. The ratio of the dynamic to static strains is the so-called dynamic response factor. Such work was extended in recent times by seeking the response of composite single skin and sandwich panels to drop load tests, which sought to simulate slam conditions, Hayman *et al.* (1991). This showed that slam pressure is not uniform on a panel; the pressure pulse typically starts at one edge of a panel and works its way to the other edge – as shown in Figure 9.25. Such work has now been incorporated into design guidelines from classification societies, Lloyd's Register of Shipping (1988b).

A further description of slamming is contained in Section 4.1.2.22.

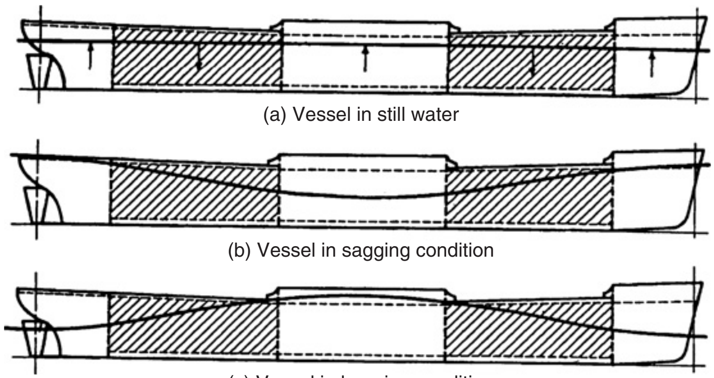


Figure 9.24 Superposition of the static wave profile.

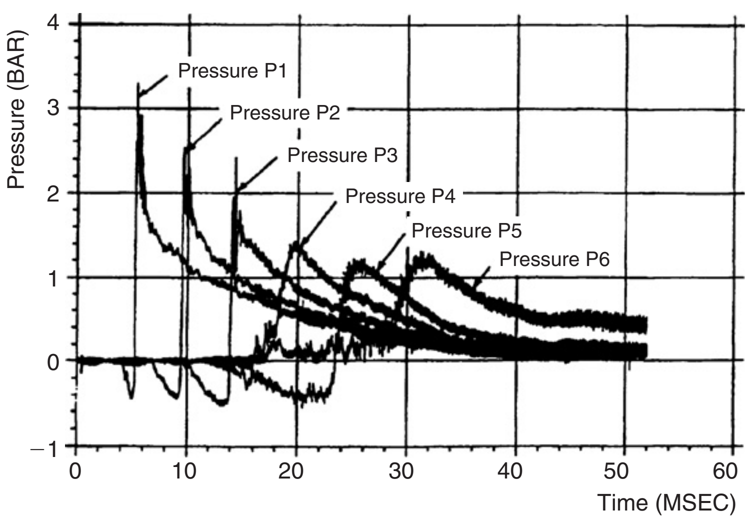


Figure 9.25 Slam pressure variation on an FRP plate panel (after Hayman *et al.* 1991).

(c) Design margins

Typical design margins based on current practice are listed in Table 9.8. The margins for short-term static loads are those which, if applied to the ultimate strength of the laminate, will give the resin microcracking stress. Stresses higher than the microcracking stress are deemed to cause significant permanent damage to the laminate, although the structure would still be able to take further load up to the ultimate value.

In case of local buckling of panels between stiffeners, the low margin of 1.5 is only justified if positive measures are taken to prevent premature detachment of stiffeners from the panel. This may involve bolting of flanges to the panel or using resilient adhesive which prevents peeling of the flanges.

Regarding fatigue, the margin of 5.0 applied to the ultimate stress only relates to high strain rate applications such as slamming and whipping in the forward regions of the ship. Static short-term margins may be used for the overall structure if the maximum operating strains are less than about 20% of the ultimate limit of the matrix.

Bearing in mind the low modulus of FRP, it is important to evaluate the structural deformations carefully. When designing tanks to withstand internal pressure, a limit of $L/200$ (where L is the tank length) is typically imposed. This avoids excessive

Table 9.8 Design margins.

Load action	*Margin*
Static short-term loads (tension)	3.0
Static short-term loads (compression)	2.0
Static long-term loads (dry)	4.0
Static long-term loads (immersed)	6.0
Load reversal	5.0
Local buckling (stiffeners parallel to load)	1.5
Column buckling of plate/stiffener combinations	2.0
Buckling (stiffeners perpendicular to load)	3.5

deformation and possible subsequent damage to boundary joints. Panels between stiffeners on lightweight decks, for example in the superstructure, should be limited in deflection to $B/80$ (where B is the panel width) to avoid them feeling springy to walk on. Careful consideration should be given to the selection of coatings (e.g., non-skid deck paint) as materials formulated for application to steel may not be sufficiently flexible for use on an FRP substrate.

9.3.4.5 *Design synthesis*

(a) Choice of topology

There are four radically different styles to choose from – top hat stiffened single skin, monocoque single skin, sandwich, and corrugated construction. The advantages and disadvantages of each of these are given in Table 9.9. From this it may be concluded that:

(i) Sandwich construction offers a fairly low cost (at least for one-offs or small production runs) and high stiffness-to-weight at the potential expense of service durability. However, these aspects are being addressed in the context of small craft and the experience should no doubt filter through to the applications in larger ships.

(ii) With a large capital investment, monocoque construction may be mechanized to a very large extent, thereby minimizing labour cost. However, the result is a heavy structure. It is best suited for long production runs and where the vessel weight is not of particular consequence. Quality assurance during build and operation can also be problematical, thereby potentially restricting use to more sophisticated customers (e.g., navies).

(iii) Stiffened single skin construction offers the lowest technical risk in that design, build inspection, maintenance, and repair are all straightforward. Thus, where weight and durability under a variety of load conditions need to be good, where weight has a slightly lower emphasis, single skin stiffened construction is suitable; this is particularly so for displacement vessels (as opposed to high-speed craft, where dynamic lift implies weight criticality). Cost is higher than for sandwich, especially for one-offs; this difference though is reduced when production runs of five or more vessels are planned.

(iv) Corrugated construction offers lighter weight than stiffened single skin, but is unlikely to be considered for hull structures without considerable further development. It is relatively expensive, particularly in terms of tooling cost and lay-up complexity.

It is possible to mix these different forms of construction to combine the advantages and obtain the best compromise for a particular application (see Table 9.10). For example, it may be attractive to specify a single skin hull and main deck, corrugated watertight bulkheads, and sandwich construction for secondary structure such as internal decks, minor bulkheads, and superstructure. The UK Sandown minehunters and the RNLI Severn class lifeboat are examples where two or more construction styles have been used.

(b) Structural elements

1. General

As mentioned in Section 9.3.4.2, ship design is characterized by the need to have a workable set of plans at very short notice. Structural design consequently suffers from constraints. The tendency of designers is to start from a known case, modify it slightly to suit changed circumstances for the new design, and then test key elements of the design in more detailed studies.

This approach is being questioned now in view of the fact that there is a growing tendency among ship owners to ask for much higher performances from the new ships. Designers are having less and less past material to base their empiricism on; increasingly therefore, designs are being based on confirmed first principles.

Structural design is based on three principal levels of load–response estimation, namely the primary (hull bending), secondary (plate bending), and tertiary (or stiffener) stresses. In general, the primary stresses are the dominant stresses for larger ships, e.g., $L > 60$ m. Even so, detail calculations are required to ensure that suitable margins exist in the structure to be able to cope with the variety and potential severity of loads.

Table 9.9 Comparison of structural styles.

Configuration	*Advantages*	*Disadvantages*
Top hat stiffened single skin	Properties and responses well known Automation possible Easy to fit equipment Costs reduce with number of hulls Quality control is easy Survey in service is straightforward	Fairly expensive to build Care is needed to provide good impact resistance
Monocoque single skin	Easily automated Low labour cost Few secondary bonds below waterline Good shock resistance	Very heavy High material cost Survey methods difficult Attachments and support to machinery difficult Quality control difficult
Sandwich	High specific bending stiffness Can be built without a mould Secondary bonding can be minimized Construction/maintenance costs low Easy to fit equipment	Survey methods need refinement Long-term durability is potential problem Precautions needed to protect core from fire
Corrugated	Relatively lightweight Low labour and material cost Automation is possible	Lower transverse strength Internal fitting may prove to be difficult Awkward mould Strange appearance

Table 9.10 Comparison of weights and costs for different structural styles.

Configuration	*Relative weight*	*Relative cost*
Single skin—longitudinal stiffening	1.00	1.00 (0.75)[a]
Corrugations with 0.16 m depth	1.24	1.55
PVC foam core sandwich	0.73	0.62
Monocoque thick GRP	3.04	1.92

[a]Compliant resin used instead of bolts.

2. Plating design

The plating thickness is determined by the requirement for it to resist a combination of lateral pressures and in-plane loading. The magnitudes and proportions of the loads vary from location to location in the ship. For instance in larger ships, the bottom shell will be subject lateral loading owing to local water from the outside, payload or cargo weight from the inside, and in-plane loading owing to global bending of the hull girder. Bulkheads are generally designed on the basis of linearly varying water pressure arising from one of the two compartments that the bulkhead separates being flooded. The deck, for instance, requires a particularly critical examination. This is because, under a sagging condition, compressive stresses in the deck plating can be significant enough to warrant exhaustive stability checks. If, in addition, there are transverse loads on the deck, then the situation becomes even more severe.

The basis of design is orthotropic plate analysis. There are several versions. At the simplest, designers use fundamental equations from isotropic theory with orthotropy incorporated by lumping the section properties on to the plate thickness to give artificially contrived panel stiffnesses, Smith (1968). A quick check can of course be made using cylindrical bending equations of the type derived by Pagano (1968). More refined approaches which incorporate shear inertia effects, Ochoa and Reddy (1992), are now being used more and more in order to specify scantlings for sandwich plates.

Failure limits in strength need to be identified explicitly. They are based on phenomenological issues:

(i) matrix cracking
(ii) fibre breakage
(iii) fibre–matrix debonding
(iv) interfacial cracking
(v) delamination
(vi) core shear cracking
(vii) skin wrinkling
(viii) skin-core debond.

This does require an explicit definition on a structural level and is done using one of the many different macroscopic criteria, e.g., maximum stress/strain, Tsai–Wu, Tsai–Hill, Hoffman, Hart-Smith, etc.

3. Stiffener design

For top hat stiffeners such as that illustrated in Figure 9.26, the designer has to select from a range of variables:

(i) section height
(ii) section width
(iii) web angle
(iv) flange width
(v) web lay-up
(vi) table lay-up.

The almost infinite freedom that this represents is tempered by the need to standardize as much as possible. The penalty for not doing so is to give the production department an almost impossible task in shaping foam former sections, tapering from one size to another, and tailoring cloth widths to suit varying section sizes. Too many changes in the number of lay-up plies along the run of one stiffener can cause many problems to the laminator. Overall the result can be a most significant reduction in productivity.

The best approach, Dodkins *et al.* (1994), is to devise a range of standard section sizes, preferably 10 or less for a ship, and try to restrict the choice to one section size for the length of each run of a stiffener. For example, the longitudinal hull bottom stiffener may run through several compartments, being supported at different, possibly unevenly spaced, locations. By selecting from the standard range, it should be possible to cope with the different spans by varying the lay-up from one compartment to the next and achieve this without incurring significant weight penalty.

The failure modes that are of interest here are:

(i) shear failure of the webs
(ii) tensile/compressive failure of the table
(iii) tensile/compressive failure of the base panel
(iv) local buckling of the table
(v) shear buckling of the webs
(vi) interlaminar shear/tensile failure of connection between flange and base plate.

4. Joints

Joints become necessary in a structure for three main reasons. These relate to production or processing restrictions, the need to gain access within the structure during its working life, and repair of the original structure.

The production-related feature arises because large structures cannot be formed in one process, thereby needing components to be joined to produce

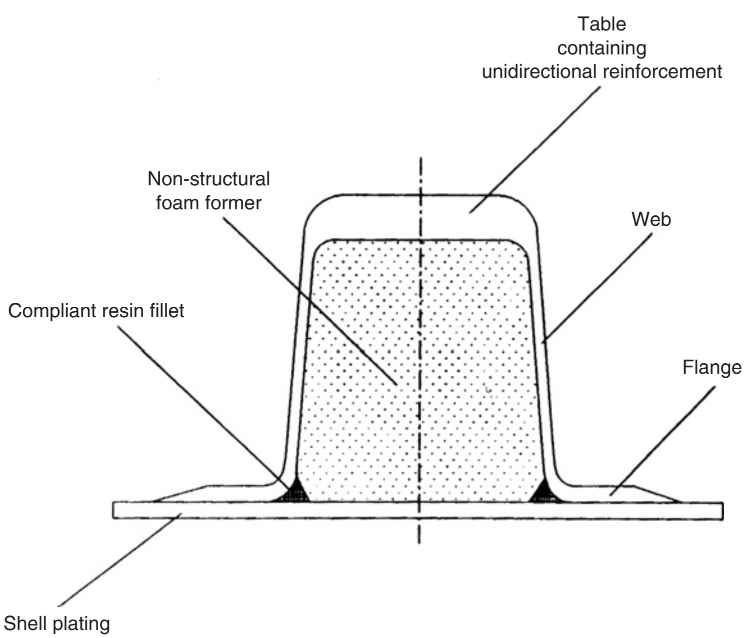


Figure 9.26 Top hat stiffener configuration.

the completed product. Considerations that limit process size include exotherm, resin working time, cloth size and drapeability, mould accessibility, and release limitations. Considering access and repair, if the components within the structure require regular servicing, then the structural elements that obstruct access need to be joined to the remaining structure in such a way as to allow them to be removed with reasonable ease. If the hidden components require only very occasional treatment, then the structure can be cut out as necessary and treated as a repair. Here the jointing method can be treated as permanent.

There are two main classes of joints, namely those that effect in-plane load transfer and those that connect two structural elements orthogonal to each other. The latter can refer either to frame-to-shell connections or bulkhead-to-shell connections.

(i) In-plane joints

These can be either bonded or bolted; the choice depends very much on the application being considered. Typical examples of bonded joints are shown in Figure 9.27. A bonded connection provides a greater area to transmit load. This ensures that all the fibres at the joint interface are used to carry load so that stress concentrations are reduced. They are cheaper and easier to produce and can be formed from one side of the panel. However, some environmental control is usually necessary during the construction process. One shortcoming is that when initial failure occurs in a purely bonded joint, it can propagate easily since there are no fibres across the joint to act as crack arrestors. Therefore special attention is devoted to the design to ensure that such events are minimized. Such joints are permanent and cannot be easily removed.

Bolted connections provide a strong link across the joint interface; they are easily removed and can usually be formed under adverse conditions. When used in conjunction with an adhesive, the bolts can act as crack arrestors in the event of final failure. However, since the load is transmitted through a small area, stress concentrations occur that can lead to early failure. They require access from both sides of the plate panels, are heavy, and can be expensive to build.

The literature on in-plane connections is extensive, Godwin and Matthews (1980), Greene (1997), and the reader can refer to such work for a clearer exposition of the subject. In the marine context, most in-plane connections between two panels are done using primarily bonded connections, Smith (1990).

(ii) Frame-to-shell connections

Some typical arrangements of such connections are shown in Figure 9.28. Frames are normally laid up over a foam former and bonded to the shell when the latter is fully cured. The main purpose of this connection is to transmit shear stresses between the shell and frame flanges under local bending caused by lateral pressure or concentrated lateral loads. Design of the connection, Greene (1997), requires an evaluation of the envelope of the maximum shear forces in each frame. Another development, Dodkins *et al.* (1994), is that of preforming top hat sections and bonding these cured sections to the shell. The major benefits of this approach are reduced production time and cost and also greater flexibility in the design of the joint.

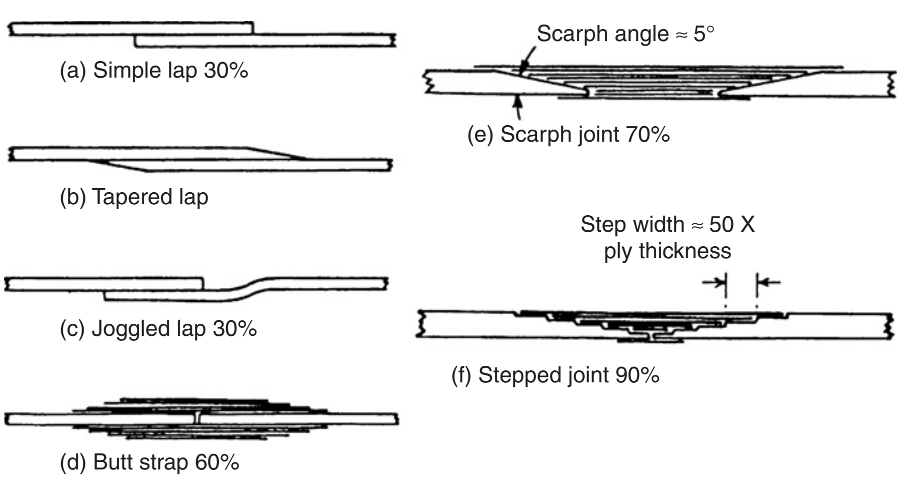


Figure 9.27 Typical arrangements and efficiencies of in-plane joints.

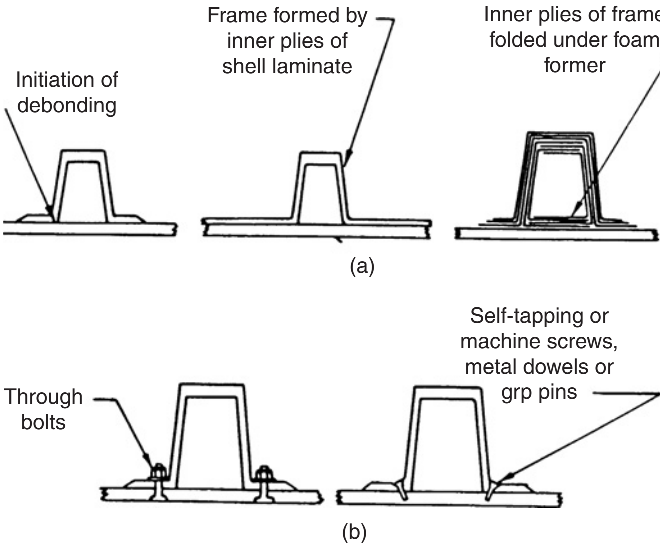


Figure 9.28 Typical frame-to-shell connections: (a) types of attachment; (b) reinforcement of joint.

(iii) Bulkhead-to-shell connections

Most ship and boat hulls rely critically on transverse bulkheads to provide rigidity and strength under transverse loads; this involves the transmission of direct and membrane shear stresses across the bulkhead-to-shell connection. An effective arrangement is provided by a double-angle arrangement; examples of such arrangements in sandwich construction are shown in Figure 9.29. Design of the boundary angle has principally been based on equating its stiffness with those of the two plates being connected, i.e., the bulkhead and shell plates. Since, in most cases, the material used in the boundary angle is the same as that in the parent plates, the thickness of the overlaminate is usually specified as a function of the thickness of the two plates. However, more recent work has shown the importance of designing joints to be flexible, Shenoi and Hawkins (1992). This is in order to avoid the effect of a 'hard point' created by the very presence of the bulkhead plate and avoid stress concentrations.

5. Finite element analysis

To model a ship's hull, or even a section, using layered finite elements would be an extremely laborious task and would require a great deal of computing resources. The preferred approach is to conduct a multilevel numerical modelling exercise. The global response of the hull structure can be modelled with sufficient accuracy using general-purpose codes and isotropic elements. This gives a reasonably realistic distribution of strains around the hull section and deformation of the hull and deck panels between bulkheads. Figure 9.30 shows a typical stress output from a study of a minehunter.

To examine stress distribution at a detailed level, local models of stiffened panels can be created and boundary conditions can be determined from the global model. Layered orthotropic elements can be used at this stage. Then all the pertinent failure mechanisms such as those listed in the previous sections can be examined. These detailed results can be used for design optimization purposes, where the lay-up can be verified and altered to yield the correct response modes without deficiency. Figure 9.31 illustrates the detail that can be obtained through modelling with this level of care.

(c) Arrangement and layout issues

1. Influence of the general arrangement

Certain features of the ship's general arrangement can have a marked influence on the complexity, and hence cost, of the structure. In particular, the following points should be noted in order to keep the structural arrangement as simple as possible.

(i) Major bulkheads should be placed in positions of multiples of frame spacing. This avoids the complication of varying frame spacing along the ship's length or landing bulkheads on the shell in positions too close to existing frames.
(ii) Bulkhead positions should lead to approximately equal compartment lengths along the length of

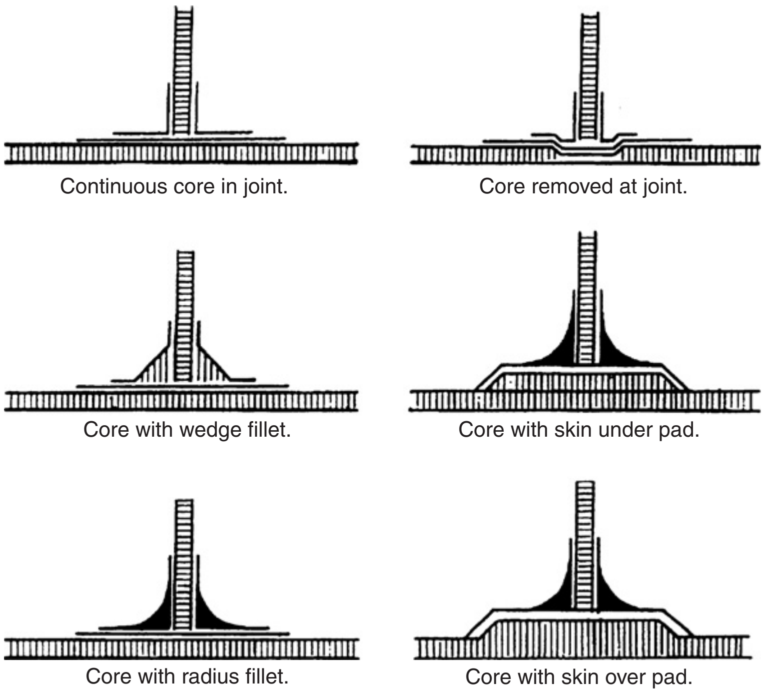


Figure 9.29 Typical tee connections in marine sandwich construction.

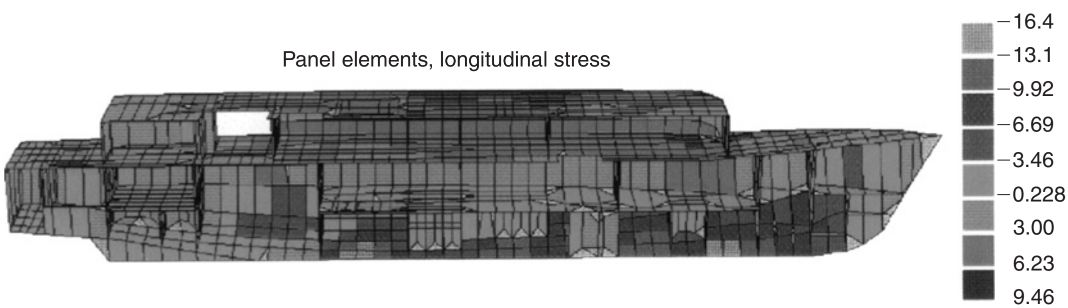


Figure 9.30 Finite element modelling of a whole ship.

the ship. This is not always practical to achieve, but in extreme cases of long compartments adjacent to short compartments, it may be necessary to taper longitudinal stiffeners, resulting in a high labour effort to shape the foam formers and tailor the lay-up cloths. It is preferable to maintain a constant former section and accommodate reasonable variations in spans by varying lay-up alone.

One exception is likely to be in the engine room space, where longitudinals have large spans, but in any case need to be shaped to provide engine and gear box foundations.

(iii) It is not essential to position main transverse bulkheads at either end of the lower tier of the superstructure. The flexibility of the FRP material will ensure that there are no significant stress concentrations at these locations.

(iv) In optimizing transverse and longitudinal frame spacing, it is important to consider the space between stiffeners required for bolted skin fittings as well as ensuring good access to all stiffener surfaces for laminators. This means a frame spacing of about 1.0–1.5 m for hull and main deck and 0.6–1.0 m for superstructure and internal structure.

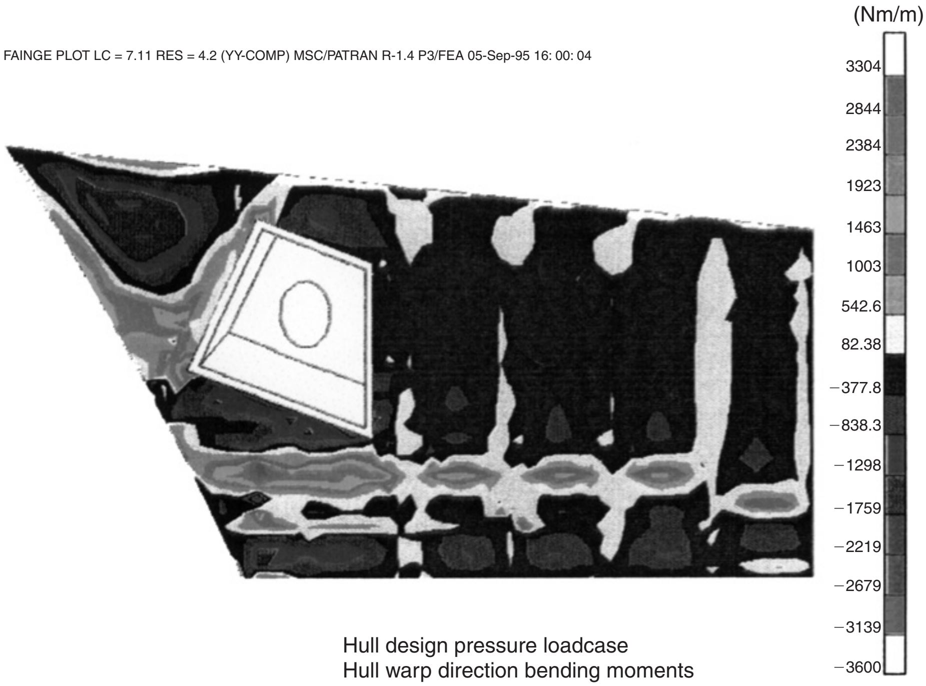


Figure 9.31 Finite element modelling of a stiffened panel.

(v) The main deck is required to have a number of hatches and shipping openings. These should be confined to the centre of the ship and kept as far apart as possible. Thus longitudinals can run straight and parallel to the centreline outside the line of openings, with transverse beams running between inner longitudinals to provide local support to the edges of the openings. This maximizes the longitudinal section modulus and avoids cranking of the longitudinals around openings (which adds to the complexity and reduces labour productivity).

(vi) In positioning the deck and bulkhead penetrations, allowance should be made for tee joints at bulkhead-to-shell and bulkhead-to-deck connections. Penetrations should be kept clear of these joints, although bonding angles can be through-bolted to provide a strong attachment point and also serve to clamp the bonding angle to the plating.

(vii) A unique property of composite materials is that almost any shape of structure may be produced by the use of an appropriately shaped mould. However, as far as possible and where feasible, efforts should be made to maximize the use of flat panels and assemblies. This is particularly so for superstructures, deck houses, and other secondary structural regions.

2. Structural arrangement

Figure 9.32 shows the structural elements in a midship section of a modern FRP vessel of predominantly stiffened single skin construction. A key feature of modern design, Dodkins (1993), is the adoption of longitudinal framing. Advantages of this form of stiffening (over transverse framing) are that:

(i) more of the structure is effective in resisting hull girder bending;
(ii) stiffener intersections are greatly reduced;
(iii) instability problems, especially in the deck structure, are minimized.

These have had to be weighted against the perceived drawbacks, which are:

(i) stiffener bases must be shaped to land upright on the varying deadrise angle of the ship's bottom;
(ii) laminating longitudinals on the side shell is difficult;
(iii) the main transverse bulkheads need to be stronger and heavier in order to support the longitudinals.

The lower ends of the side frames are simply butted onto the outermost bottom longitudinals at the turn of the bilge. This part of the hull structure is inherently rigid and external pressures do not place excessive

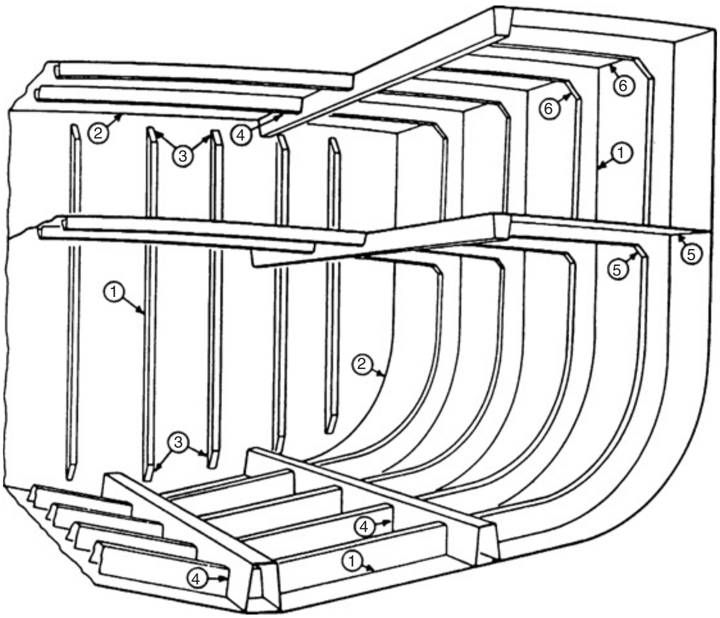


Figure 9.32 Midship section of a Sandown class minehunter.

load on these joints. The upper ends of the joints are terminated alternately by a snape or a beam knee connection to the main deck beams. The arrangement ensures good continuity and transverse strength between the hull and the main deck structure.

Deck plating thickness ranges from 15 to 25 mm, reflecting the variation of longitudinal bending and demands of local loading. Shell plating is about 20 mm thick in most parts, with extra reinforcement placed locally by way of highly loaded regions such as the forward end which is prone to slamming loads, tanks where there is local fluid loading, and in the engine room where extra stiffness has to be provided for machinery supports.

9.3.4.6 *External issues*

(a) Regulatory issues

Two features that characterize ships, which have been alluded to in Section 9.3.4.2, and which influence design practice, are the very short lead time from tender/order to delivery of the ships and the fact that they are generally made-to-order, one-off products. The effect of these is to place tremendous pressure on designers to produce designs that are both practical and optimal. An ideal approach to adopt would be one based entirely on first principles. In such practice, all possible combinations of requirements would need to be assessed thoroughly and hypotheses tested rigorously. Such assessment would be time-consuming and expensive – two luxuries that are ill-affordable by the marine community. Primarily because of this set of constraints, designers place a great deal of reliance on ‘rules and regulations’ of respected independent regulatory bodies – classification societies such as Lloyd’s Register of Shipping, the American Bureau of Shipping, Det Norske Veritas, Germanischer Lloyd, Bureau Veritas. See also Section 11.3.

Design codes are the instruments through which classification societies exercise a partial control over the design activity. Codes specify minimum requirements to be satisfied by any designer. The adoption of optimal solutions is a natural attitude of a rational designer, which leads very often to a design based on minimum code requirements. Thus codes govern the main features of design and they also represent the existing practice in a sector of the marine industry. They result from the experience of applying evolving guidance principles and they shape the new ships to be produced. Because the codes need to be universal in application, both geographically and in terms of the product range, they have to be simple to use. This, in turn, implies that the expressions used to calculate the design variables and parameters have to be simple to understand and apply. The simplicity sometimes conflicts with the need to assure adequate safety margins. This forces the classification societies to be quite conservative to balance the lack of accuracy in the design formulations.

Very recent developments in information technology and the proper harnessing of computing power however are encouraging. This is allowing an integration of hydrodynamics and loading calculations, definition of ship geometry, synthesis of structural elements, materials characterization, and production modelling capabilities. Designers are thus being able to assess the global effects of the change of a structural design parameter on whole ship performance fairly quickly. The capabilities in this context are still in their infancy. Regulatory bodies and the insurance industry that underwrites the financing of ships and shipping need to be convinced of the validity and correctness of such tools. The validation process is underway in a number of different ways and forums. The entirely first principles based process should therefore be a reality soon.

(b) Statutory issues

Apart from the issues discussed above, all ships have to conform to statutes of the country in which they are registered. These laws are, in the main, derived from resolutions of the International Maritime Organization (IMO), see Section 11.2.2.

The principal relevance of the IMO and the statutory implications in ship structural design is that there is a requirement for the main structure in ships to be built of non-combustible materials. Steel is a non-combustible material; aluminium alloys (even though they melt at relatively low temperatures) also do not burn. Both these are acceptable structural materials for ships. FRP composites, however, are combustible. Therefore they are subject to stringent checks under various clauses. The most recent example of this is the adoption of a code for the design of high-speed craft, IMO (2000), HSC code.

The HSC code applies to vessels of high speed which are engaged in international voyages, covering passenger craft which do not proceed more than four hours from a port of refuge, and cargo craft of 500 gross tonnes and upwards, which do not proceed for more than eight hours from a port of refuge. The HSC code includes requirements of 'fire restrictive' (or combustible) materials with respect to their use in primary, secondary, and tertiary structures and components. The requirements of the HSC code are principally aimed at:

(i) *fire prevention* – the use of non-combustible or fire-restricting materials, such that fire prevention is controlled by low flame spread materials, limited heat flux and limited heat release, together with the control of harmful gases and smoke;
(ii) *structural performance* – controlling the structural integrity at elevated temperatures;
(iii) *fire containment* – controlling fires developed in major and moderate fire hazard areas by the use of fire resisting divisions.

These place tremendous burdens on the designer to demonstrate conformance of the structure with the HSC requirements. Current attributes of the candidate materials are such that FRP composites will rarely be allowed for use in structural applications in ships. However, the code and requirements are being re-examined in a more fundamental manner both in terms of evaluation of the safety case for ships where the whole picture of passenger safety and evacuation following a fire is considered (rather than one of just the candidate materials) and in terms of prescribing the correct tests for checking conformance. In a curious and paradoxical context, FRP composites are being used in offshore structures following the disastrous Piper Alpha fire precisely because of their fire-resistive capabilities, Gibson (1993). The future therefore looks promising for the application of polymeric composites in major ship structural applications.

9.3.5 Corrosion

9.3.5.1 *Nature and forms of corrosion*

There is a natural tendency for nearly all metals to react with their environment. The result of this reaction is the creation of a corrosion product which is generally a substance of very similar chemical composition to the original mineral from which the metal was produced.

Atmospheric corrosion. Protection against atmospheric corrosion is important during the construction of a ship, both on the building berth and in the shops. Serious rusting may occur where the relative humidity is above about 70%; the atmosphere in British shipyards is unfortunately sufficiently humid to permit atmospheric corrosion throughout most of the year. But even in humid atmospheres the rate of rusting is determined mainly by the pollution of the air through smoke and/or sea salts.

Corrosion due to immersion. When a ship is in service the bottom area is completely immersed and the waterline or boot topping region may be intermittently immersed in sea water. Under normal operating conditions a great deal of care is required to prevent excessive corrosion of these portions of the hull. A steel hull in this environment can provide ideal conditions for the formation of electro-chemical corrosion cells.

Electro-chemical nature of corrosion. Any metal in tending to revert to its original mineral state releases energy. At ordinary temperatures in aqueous solutions the transformation of a metal atom into a mineral molecule occurs by the metal passing into solution. During this process the atom loses one or more electrons and becomes an ion, i.e. an electrically charged atom, with the production of an electric current (the released energy). This reaction may only occur if an electron acceptor is present in the aqueous solution. Thus any corrosion reaction is always accompanied by a flow of electricity from one metallic area to another through a solution in which the conduction of an electric current occurs by the passage of ions. Such a solution is referred to as an electrolyte solution; and because of its high salt content sea water is a good electrolyte solution.

A simple corrosion cell is formed by two different metals in an electrolyte solution (a galvanic cell) as illustrated in Figure 9.33. It is not essential to have two different metals as we shall see later. As illustrated a pure iron plate and a similar pure copper plate are immersed in a sodium chloride solution which is in contact with oxygen at the surface. Without any connection the corrosion reaction on each plate would be small. Once the two plates are connected externally to form an electrical path then the corrosion rate of the iron will increase considerably, and the corrosion on the copper will cease. The iron electrode by means of which the electrons leave the cell and by way of which the conventional current enters the cell is the anode. This is the electrode at which the oxidation or corrosion normally takes place. The copper electrode by means of which the electrons enter the cell and by way of which the conventional current leaves the cell is the cathode, at which no corrosion occurs. A passage of current through the electrolyte solution is by means of a flow of negative ions to the anode and a flow of positive ions to the cathode.

Electro-chemical corrosion in aqueous solutions will result from any anodic and cathodic areas coupled in the solution whether they are metals of different potential in the environment or they possess different potentials as the result of physical differences on the metal surface. The latter is typified by steel plate carrying broken millscale in sea water (Figure 9.33) or corrosion currents flowing between areas of well painted plate and areas of defective paintwork.

In atmospheric corrosion and corrosion involving immersion both oxygen and an electrolyte play an important part. Plates freely exposed to the atmosphere will receive plenty of oxygen but little moisture, and the moisture present therefore becomes the controlling factor. Under conditions of total immersion it is the presence of oxygen which becomes the controlling factor.

Bimetallic (galvanic) corrosion. Although it is true to say that all corrosion is basically galvanic, the term 'galvanic corrosion' is usually applied when two different metals form a corrosion cell.

Many ship corrosion problems are associated with the coupling of metallic parts of different potential which consequently form corrosion cells under service conditions. The corrosion rates of metals and alloys in sea water have been extensively investigated and as a result galvanic series of metals and alloys in sea water have been obtained.

A typical galvanic series in sea water is shown in Table 9.11.

The positions of the metals in the table apply only in a sea water environment; and where metals are grouped together they have no strong tendency to form couples with each other. Some metals appear twice because they are capable of having both a passive

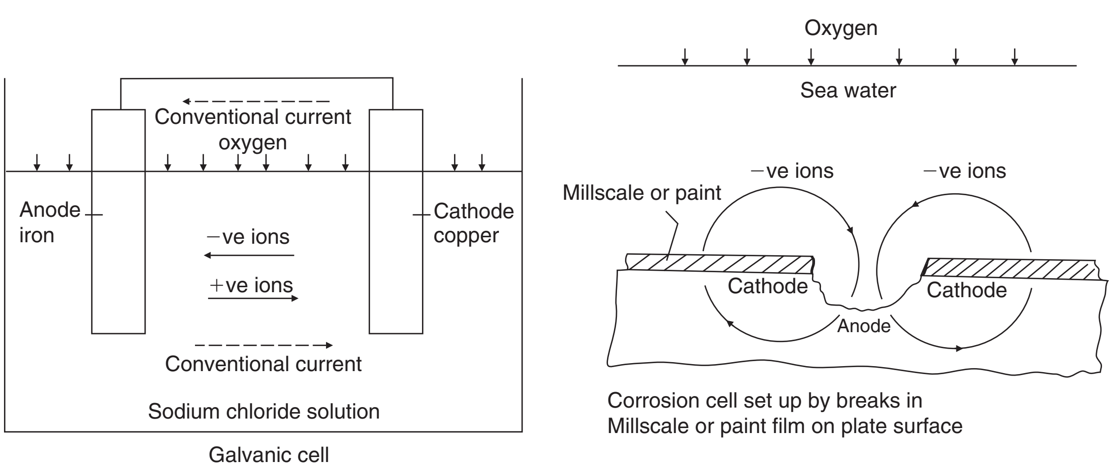


Figure 9.33 Corrosion cell.

Table 9.11 Galvanic series of metals and alloys in sea water.

Noble (cathodic or protected) end
Platinum, gold
Silver
Titanium
Stainless steels, passive
Nickel, passive
High duty bronzes
Copper
Nickel, active
Millscale
Naval brass
Lead, tin
Stainless steels, active
Iron, steel, cast iron
Aluminium alloys
Aluminium
Zinc
Magnesium
Ignoble (anodic or corroding) end

and an active state. A metal is said to be passive when the surface is exposed to an electrolyte solution and a reaction is expected but the metal shows no sign of corrosion. It is generally agreed that passivation results from the formation of a current barrier on the metal surface, usually in the form of an oxide film. This thin protective film forms, and a change in the overall potential of the metal occurs when a critical current density is exceeded at the anodes of the local corrosion cells on the metal surface.

Among the more common bimetallic corrosion cell problems in ship hulls are those formed by the mild steel hull with the bronze or nickel alloy propeller. Also above the waterline problems exist with the attachment of bronze and aluminium alloy fittings. Where aluminium superstructures are introduced, the attachment to the steel hull and the fitting of steel equipment to the superstructure require special attention. This latter problem is overcome by insulating the two metals and preventing the ingress of water as illustrated in Figure 9.34. A further development is the use of explosion-bonded aluminium/steel transition joints also illustrated. These joints are free of any crevices, the exposed aluminium to steel interface being readily protected by paint.

Stress corrosion. Corrosion and subsequent failure associated with varying forms of applied stress is not uncommon in marine structures. Internal stresses produced by non-uniform cold working are often more dangerous than applied stresses. For example, localized corrosion is often evident at cold flanged brackets.

Corrosion/erosion. Erosion is essentially a mechanical action but it is associated with electro-chemical corrosion in producing two forms of metal deterioration. Firstly, in what is known as 'impingement attack' the action is mainly electro-chemical but it is initiated by erosion. Air bubbles entrained in the flow of water and striking a metal surface may erode away any protective film that may be present locally. The eroded surface becomes anodic to the surrounding surface and corrosion occurs. This type of attack can occur in most places where there is water flow, but particularly where features give rise to turbulent flow. Sea water

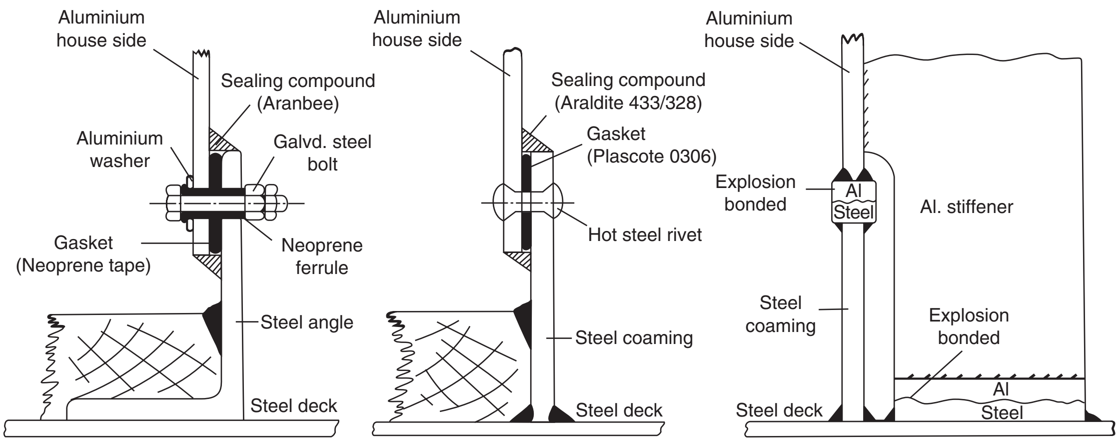


Figure 9.34 Aluminium to steel connections.

discharges from the hull are a particular case, the effects being worse if warm water is discharged.

Cavitation damage is also associated with a rapidly flowing liquid environment. At certain regions in the flow (often associated with a velocity increase resulting from a contraction of the flow stream) the local pressures drop below that of the absolute vapour pressure. Vapour cavities, that is areas of partial vacuum, are formed locally, but when the pressure increases clear of this region the vapour cavities collapse or ‘implode’. This collapse occurs with the release of considerable energy, and if it occurs adjacent to a metal surface damage results. The damage shows itself as pitting which is thought to be predominantly due to the effects of the mechanical damage. However it is also considered that electro-chemical action may play some part in the damage after the initial erosion. See also Section 5.5.

Corrosion allowance. Plate and section scantlings specified for ships in the rules of classification societies include corrosion additions to the thickness generally based on a 25 year service life. The corrosion allowance is based on the concept that corrosion occurs on the exposed surface of the material at a constant rate, no matter how much material lies behind it. That is if a plate is 8 mm or 80 mm thick, corrosion will take place at the same rate, not at a faster rate in the thicker plate.

9.3.5.2 *Corrosion control*

The control of corrosion may be broadly considered in two forms, cathodic protection and the application of protective coatings, i.e. paints.

Cathodic protection. Only where metals are immersed in an electrolyte can the possible onset of corrosion be prevented by cathodic protection. The fundamental principle of cathodic protection is that the anodic corrosion reactions are suppressed by the application of an opposing current. This superimposed direct electric current enters the metal at every point lowering the potential of the anode metal of the local corrosion cells so that they become cathodes.

There are two main types of cathodic protection installation, sacrificial anode systems and impressed current systems.

(1) *Sacrificial anode systems* – Sacrificial anodes are metals or alloys attached to the hull which have a more anodic, i.e. less noble, potential than steel when immersed in sea water. These anodes supply the cathodic protection current, but will be consumed in doing so and therefore require replacement for the protection to be maintained.

    This system has been used for many years, the fitting of zinc plates in way of bronze propellers and other immersed fittings being common practice. Initially results with zinc anodes were not always very effective owing to the use of unsuitable zinc alloys. Modern anodes are based on alloys of zinc, aluminium, or magnesium which have undergone many tests to examine their suitability; high purity zinc anodes are also used. The cost, with various other practical considerations, may decide which type is to be fitted.

    Sacrificial anodes may be fitted within the hull, and are often fitted in ballast tanks. However, magnesium anodes are not used in the cargo-ballast tanks of oil carriers owing to the ‘spark hazard’. Should any part of the anode fall and strike the tank structure when gaseous conditions exist an explosion could result. Aluminium anode systems may be employed in tankers provided they are only fitted in locations where the potential energy is less than 28 kg.m.

(2) *Impressed current systems* – These systems are applicable to the protection of the immersed external hull only. The principle of the systems is that a voltage difference is maintained between the hull and fitted anodes, which will protect the hull against corrosion, but not overprotect it thus wasting current. For normal operating conditions the potential difference is maintained by means of an externally mounted silver/silver chloride reference cell detecting the voltage difference between itself and the hull. An amplifier controller is used to amplify the micro-range reference cell current, and it compares this with the preset protective potential value which is to be maintained. Using the amplified DC signal from the controller a saturable reactor controls a larger current from the ship’s electrical system which is supplied to the hull anodes. An AC current from the electrical system would be rectified before distribution to the anodes. Figure 9.35 shows such a system.

Originally, consumable anodes were employed but in recent systems non-consumable relatively noble metals are used; these include lead/silver and platinum/palladium alloys, and platinized titanium anodes are also used.

A similar impressed current system employs a consumable anode in the form of an aluminium wire up to 45 metres long which is trailed behind the ship whilst at sea. No protection is provided in port.

Although the initial cost is high, these systems are claimed to be more flexible, to have a longer life, to reduce significantly hull maintenance, and to weigh less than the sacrificial anode systems.

Care is required in their use in port alongside ships or other unprotected steel structures.

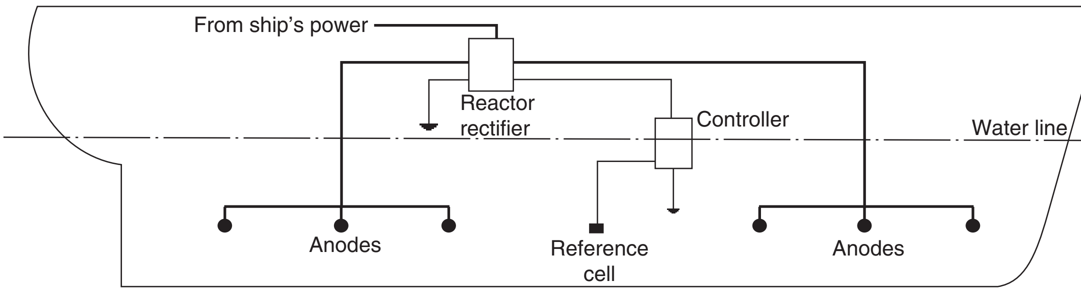


Figure 9.35 Impressed current cathode protection system.

Protective coatings (paints). Paints intended to protect against corrosion consist of pigment dispersed in a liquid referred to as the 'vehicle'. When spread out thinly the vehicle changes in time to an adherent dry film. The drying may take place through one of the following processes.

(a) When the vehicle consists of solid resinous material dissolved in a volatile solvent, the latter evaporates after application of the paint, leaving a dry film.
(b) A liquid like linseed oil as a constituent of the vehicle may produce a dry paint film by reacting chemically with the surrounding air.
(c) A chemical reaction may occur between the constituents of the vehicle after application, to produce a dry paint film. The reactive ingredients may be separated in two containers ('two-pack paints') and mixed before application. Alternatively ingredients which only react at higher temperatures may be selected, or the reactants may be diluted with a solvent so that the reaction occurs only slowly in the can.

Corrosion-inhibiting paints for application to steel have the following vehicle types:

(a) *Bitumen or pitch* Simple solutions of bitumen or pitch are available in solvent naphtha or white spirit. The bitumen or pitch may also be blended by heat with other materials to form a vehicle.
(b) *Oil based* These consist mainly of vegetable drying oils, such as linseed oil and tung oil. To accelerate the drying by the natural reaction with oxygen, driers are added.
(c) *Oleo-resinous* The vehicle incorporates natural or artificial resins into drying oils. Some of these resins may react with the oil to give a faster drying vehicle. Other resins do not react with the oil but heat is applied to dissolve the resin and cause the oil to body.
(d) *Alkyd resin* These vehicles provide a further improvement in the drying time and film forming properties of drying oils. The name alkyd arises from the ingredients, alcohols and acids. Alkyds need not be made from oil, as an oil-fatty acid or an oil-free acid may be used.

(*Note.* Vehicle types (b) and (d) are not suitable for underwater service, and only certain kinds of (c) are suitable for such service.)

(e) *Chemical-resistant* Vehicles of this type show extremely good resistance to severe conditions of exposure. As any number of important vehicle types come under this general heading these are dealt with individually.
   (i) *Epoxy resins* Chemicals which may be produced from petroleum and natural gas are the source of epoxy resins. These paints have very good adhesion, apart from their excellent chemical resistance. They may also have good flexibility and toughness where co-reacting resins are introduced. Epoxy resins are expensive owing to the removal of unwanted side products during their manufacture, and the gloss finish may tend to 'chalk' making it unsuitable for many external decorative finishes. These paints often consist of a 'two-pack' formulation, a solution of epoxy resin together with a solution of cold curing agent, such as an amine or a polyamide resin, being mixed prior to application. The mixed paint has a relatively slow curing rate at temperatures below 10°C. Epoxy resin paints should not be confused with epoxy-ester paints which are unsuitable for underwater use. Epoxy-ester paints can be considered as alkyd equivalents, as they are usually made with epoxy resins and oil-fatty acids.
   (ii) *Coal tar/epoxy resin* This vehicle type is similar to the epoxy resin vehicle except that, as a two-pack product, a grade of coal tar pitch is blended with the resin. A formulation of this type combines to some extent the chemical resistance of the epoxy resin with the impermeability of coal tar.
   (iii) *Chlorinated rubber and isomerized rubber* The vehicle in this case consists of a solution of plasticized chlorinated rubber,

or isomerized rubber. Isomerized rubber is produced chemically from natural rubber, and it has the same chemical composition but a different molecular structure. Both these derivatives of natural rubber have a wide range of solubility in organic solvents, and so allow a vehicle of higher solid content. On drying, the film thickness is greater than would be obtained if natural rubber were used. High build coatings of this type are available, thickening or thixotropic agents being added to produce a paint which can be applied in much thicker coats. Coats of this type are particularly resistant to attack from acids and alkalis.

(iv) *Polyurethane resins* A reaction between isocyanates and hydroxyl-containing compounds produces 'urethane' and this reaction has been adapted to produce polymeric compounds from which paint film, fibres, and adhesives may be obtained. Paint films so produced have received considerable attention in recent years, and since there is a variety of isocyanate reactions, both one-pack and two-pack polyurethane paints are available. These paints have many good properties; toughness, hardness, gloss, abrasion resistance, as well as chemical and weather resistance. Polyurethanes are not used under water on steel ships, only on superstructures, etc., but they are very popular on yachts where their good gloss is appreciated.

(v) *Vinyl resins* Vinyl resins are obtained by the polymerization of organic compounds containing the vinyl group. The solids content of these paints is low; therefore the dry film is thin, and more coats are required than for most paints. As vinyl resin paints have poor adhesion to bare steel surfaces they are generally applied over a pretreatment primer. Vinyl paint systems are among the most effective for the underwater protection of steel.

(f) *Zinc-rich paints* Paints containing metallic zinc as a pigment in sufficient quantity to ensure electrical conductivity through the dry paint film to the steel are capable of protecting the steel cathodically. The pigment content of the dry paint film should be greater than 90%, the vehicle being an epoxy resin, chlorinated rubber, or similar medium.

Corrosion protection by means of paints. It is often assumed that all paint coatings prevent attack on the metal covered simply by excluding the corrosive agency, whether air or water. This is often the main and sometimes only form of protection; however there are many paints which afford protection even though they present a porous surface or contain various discontinuities.

For example certain pigments in paints confer protection on steel even where it is exposed at a discontinuity. If the reactions at the anode and cathode of the corrosion cell which form positive and negative ions respectively, are inhibited, protection is afforded. Good examples of pigments of this type are red lead and zinc chromate, red lead being an anodic inhibitor, and zinc chromate a cathodic inhibitor. A second mode of protection occurs at gaps where the paint is richly pigmented with a metal anodic to the basis metal. Zinc dust is a commercially available pigment which fulfils this requirement for coating steel in a salt water environment. The zinc dust is the sacrificial anode with respect to the steel.

9.3.5.3 *Anti-fouling systems*

The immersed hull and fittings of a ship at sea, particularly in coastal waters, are subject to algae, barnacle, mussel and other shellfish growth that can impair its hydrodynamic performance and adversely affect the service of the immersed fittings.

Fittings such as cooling water intake systems are often protected by impressed current anti-fouling systems and immersed hulls today are finished with very effective self polishing anti-fouling paints.

Impressed current anti-fouling systems. The functional principle of these systems is the establishment of an artificially triggered voltage difference between copper anodes and the integrated steel plate cathodes. This causes a minor electrical current to flow from the copper anodes, so that they are dissolved to a certain degree. A control unit makes sure that the anodes add the required minimum amount of copper particles to the sea water, thus ensuring the formation of copper oxide that creates ambient conditions precluding local fouling. A control unit can be connected to the management system of the vessel. Using information from the management system the impressed current anti-fouling system can determine the amount of copper that needs to be dissolved to give optimum performance with minimum wastage of the anodes.

Anti-fouling paints. Anti-fouling paints consist of a vehicle with pigments which give body and colour together with materials toxic to marine vegetable and animal growth. Copper is the best known toxin used in traditional anti-fouling paints.

To prolong the useful life of the paint the toxic compounds must dissolve slowly in sea water. Once the release rate falls below a level necessary to prevent settlement of marine organisms the anti-fouling composition is no longer effective. On merchant ships the effective period for traditional compositions was about 12 months. Demands in particular from large tanker owners wishing to reduce very high docking costs led to specially developed anti-fouling compositions with an effective life up to 24 months in the early 1970s. Subsequent developments of constant emission organic toxin antifoulings having a leaching rate independent of exposure time saw the paint technologists by chance discover coatings which also tended to become smoother in service. These so called self-polishing antifoulings with a lifetime that is proportional to applied thickness and therefore theoretically unlimited, smooth rather than roughen with time and result in reduced friction drag. Though more expensive than their traditional counterparts, given the claim that each 10 micron (10^{-3}mm) increase in hull roughness can result in a 1% increase in fuel consumption, their self polishing characteristic as well as their longer effective life, up to 5 years protection between drydockings, made them attractive to the shipowner.

The benefits of the first widely used SPC (self polishing copolymer) anti-fouling paints could be traced to the properties of their prime ingredients the tributylen compounds or TBT's. TBT's were extremely active against a wide range of fouling organisms, also they were able to be chemically bonded to the acrylic backbone of the paint system. When immersed in sea water a specific chemical reaction took place which cleaved the TBT from the paint backbone, resulting in both controlled release of the TBT and controlled disappearance or polishing of the paint film. Unfortunately, it was found that the small concentrations of TBT's released, particularly in enclosed coastal waters, had a harmful effect on certain marine organisms. This led to he banning of TBT anti-fouling paints for pleasure boats and smaller commercial ships in many developed countries and the introduction of regulations limiting the release rate of TBT for antifouling paints on larger ships. The International Convention On The Control Of Harmful Anti-Fouling On Ships, 2001 subsequently required that

(a) ships shall not apply or reapply organotin compounds which act as biocides in anti-fouling systems on or after 1 January 2003; and
(b) no ship shall have organotin compounds which act as biocides in anti-fouling systems (except floating platforms, FSU's and FPSO's built before 2003 and not docked since before 2003)

(Note! Organotin means an organic compound with one or more tin atoms in its molecules used as a pesticide, hitherto considered to decompose safely, now found to be toxic in the food chain. A biocide is a chemical capable of killing living organisms.)

Anti-fouling paints subsequently applied have generally focused on either the use of copper-based self polishing anti-fouling products, which operate in a similar manner to the banned TBT products, or the use of the so-called low-surface-energy coatings. The latter coatings do not polish or contain booster biocides, instead they offer a very smooth, low-surface-energy surface to which it is difficult for fouling to adhere. When the vessel is at rest some fouling may occur but once it is underway and reaches a critical speed the fouling is released.

9.3.5.4 *Painting ships*

To obtain the optimum performance from paints it is important that the metal surfaces are properly prepared before application of paints and subsequently maintained as such throughout the fabrication and erection process. Paints tailored for the service conditions of the structure to which they apply, and recommended as such by the manufacturer, only should be applied.

Surface preparation. Good surface preparation is essential to successful painting, the primary cause of many paint failures being the inadequacy of the initial material preparation.

It is particularly important before painting new steel that any millscale should be removed. Millscale is a thin layer of iron oxides which forms on the steel surface during hot rolling of the plates and sections. Not only does the non-uniform millscale set up corrosion cells as illustrated previously, but it may also come away from the surface removing any paint film applied over it.

The most common methods employed to prepare steel surfaces for painting are:

Blast cleaning
Pickling
Flame cleaning
Preparation by hand

(a) *Blast cleaning* is the most efficient method for preparing the surface and is in common use in all large shipyards. Following the blast cleaning it is desirable to brush the surface, and apply a coat of priming paint as soon as possible since the metal is liable to rust rapidly.

There are two main types of blasting equipment available, an impeller wheel plant where the abrasive is thrown at high velocity against the metal surface, and a nozzle type where a jet of abrasive impinges on the metal surface. The latter type should preferably be fitted with vacuum recovery equipment, rather than allow the spent abrasive and dust to be discharged to atmosphere, as is often the case in ship repair work. Impeller wheel plants which are self-contained and collect the dust and re-circulate the clean abrasive are generally fitted within the shipbuilding shops.

Cast iron and steel grit, or steel shot which is preferred, may be used for the abrasive, but non-metallic abrasives are also available. The use of sand is prohibited in the United Kingdom because the fine dust produced may cause silicosis.

(b) *Pickling* involves the immersion of the metal in an acid solution, usually hydrochloric or sulphuric acid in order to remove the millscale and rust from the surface. After immersion in these acids the metal will require a thorough hot water rinse. It is preferable that the treatment is followed by application of a priming coat.

(c) Using an *oxy-acetylene flame* the millscale and rust may be removed from a steel surface. The process does not entirely remove the millscale and rust, but it can be quite useful for cleaning plates under inclement weather conditions, the flame drying out the plate.

(d) *Hand cleaning* by various forms of wire brush is often not very satisfactory, and would only be used where the millscale has been loosened by weathering, i.e. exposure to atmosphere over a long period.

Blast cleaning is preferred for best results and economy in shipbuilding, and it is essential prior to application of high performance paint systems used today. Pickling which also gives good results can be expensive and less applicable to production schemes; flame cleaning is much less effective; and hand cleaning gives the worst results.

Temporary paint protection during building. After the steel is blast cleaned it may be several months before it is built into the ship and finally painted. It is desirable to protect the material against rusting in this period as the final paint will offer the best protection when applied over perfectly clean steel.

The formulation of a prefabrication primer for immediate application after blasting must meet a number of requirements. It should dry rapidly to permit handling of the plates within a few minutes and working the plates within a day or so. It should be non-toxic, and it should not produce harmful porosity in welds nor give off obnoxious fumes during welding or cutting. It must also be compatible with any subsequent paint finishes to be applied. Satisfactory formulations are available, for example a primer consisting of zinc dust in an epoxy resin.

Paint systems on ships. The paint system applied to any part of a ship will be dictated by the environment to which that part of the structure is exposed. Traditionally the painting of the external ship structure was divided into three regions.

(a) Below the water-line where the plates are continually immersed in sea water.
(b) The water-line or boot topping region where immersion is intermittent and a lot of abrasion occurs.
(c) The topsides and superstructure exposed to an atmosphere laden with salt spray, and subject to damage through cargo handling.

However, now that tougher paints are used for the ship's bottom the distinction between regions need not be so well defined, one scheme covering the bottom and water-line regions.

Internally by far the greatest problem is the provision of coatings for various liquid cargo and salt water ballast tanks.

(a) *Below the Water-line* The ship's bottom has priming coats of corrosion-inhibiting paint applied which are followed by an anti-fouling paint. Paints used for steels immersed in sea water are required to resist alkaline conditions. The reason for this is that an iron alloy immersed in a sodium chloride solution having the necessary supply of dissolved oxygen gives rise to corrosion cells with caustic soda produced at the cathodes. Further the paint should have a good electrical resistance so that the flow of corrosion currents between the steel and sea water is limited. These requirements make the standard non-marine structural steel primer red lead in linseed oil unsuitable for ship use below the water-line. Suitable corrosion-inhibiting paints for ships' bottoms are pitch or bitumen types, chlorinated rubber, coal tar/epoxy resin, or vinyl resin paints. The anti-fouling paints may be applied after the corrosion-inhibiting coatings and should not come into direct contact with the steel hull, since the toxic compounds present may cause corrosion.

(b) *Water-line or boot topping region* Generally modern practice requires a complete paint system for the hull above the water-line. This may be based on vinyl and alkyd resins or on polyurethane resin paints.

(c) *Superstructures* Red lead or zinc chromate based primers are commonly used. White finishing paints are then used extensively for superstructures. These are usually oleo-resinous or alkyd paints which may be based on 'non-yellowing' oils, linseed oil-based paints which yellow on exposure being avoided on modern ships.

Where aluminium alloy superstructures are fitted, under no circumstance should lead based paints be applied; zinc chromate paints are generally supplied for application to aluminium.

Cargo and ballast tanks. Severe corrosion may occur in a ship's cargo tanks as the combined result of carrying liquid cargoes and sea water ballast, with warm or cold sea water cleaning between voyages. This is particularly true of oil tankers. Tankers carrying 'white oil' cargoes suffer more general corrosion than those carrying crude oils which deposit a film on the tank surface providing some protection against corrosion. The latter type may however experience severe local pitting corrosion due to the non-uniformity of the deposited film, and subsequent corrosion of any bare plate when sea water ballast is carried. Epoxy resin paints are used extensively within these tanks, and vinyl resins and zinc rich coatings may also be used.

Further useful information on paints and anti-fouling systems is given in Anon. (2003, 2005), IMO (2005) and Swain *et al.* (2007).

9.4 Ship construction

9.4.1 Introduction

This section outlines typical examples of ship structure, and the complexity of stiffening arrangements. An outline of shipyard layout and shipbuilding process is given, together with a description of the links between the design, drawing and manufacturing process.

9.4.2 Typical examples of structure

Figures 9.36 to 9.41 illustrate some typical components of structure. Figure 9.36 shows a typical transom stern, stern frame and the stiffening arrangement in the aft peak. Figure 9.37 shows a typical midship section for a transversely framed cargo ship and Figure 9.38 the midship section for a container ship, showing side shell, bottom shell and tank top plating and stiffening arrangements. Figure 9.39 shows the midship section for a longitudinally stiffened high-speed catamaran ferry, using aluminium alloy. Double bottom construction is illustrated in Figures 9.40(a) and 9.40(b), (a) showing a transversely framed double bottom and (b) a longitudinally framed version. Figure 9.41 illustrates a fore end layout, showing the bulbous bow and fore peak structure.

9.4.3 Shipyard layout

The past two or three decades have seen the emergence of a substantial number of new shipyards, primarily in Asia and Eastern Europe, that have been specifically planned to construct the larger ships being ordered today, using contemporary shipbuilding practices and production methods. A number of traditional shipbuilders have also established new yards where they can also build larger ships and/or exploit the new technology and production methods. In general the remaining shipbuilders will have had to re-configure their site in order to utilize new technology and improve production, whilst continuing to build ships. In many cases the latter will still be restrained as to the size and type of ship that can be built.

An ideal layout for a modern shipyard is based on a production flow basis, with the yard extending back from the river or shore at which the berths or building dock are located. The furthest area from the berths is reserved for the material stockyard, and between the two are arranged in sequence the consecutive work and shop processes. Too often existing shipyards follow the river bank, and are restricted by their location in a built up area or the physical river bank slope from extending back from the river, so that modified production flow lines are required.

Planning a new shipyard, or re-planning an existing one, will involve decisions to be made on the following:

Size and type of ship to be built.
Material production per year to be achieved.
Material handling equipment to be supplied.
Machining processes to be installed.
Unit size and weight to be fabricated and erected.
Amount of outfit and engine installation to be undertaken.
Control services to be supplied.
Administration facilities required.

Shipyards usually have a fitting out basin or berth where the virtually completed ship is tied up after launching and the finishing off work and static trails may be carried out.

Before considering the actual layout of the shipyard it is as well to consider the relationship of the work processes involved in building a ship as illustrated in Figure 9.42.

An idealized layout of a new shipyard is indicated in Figure 9.43 which might be appropriate for a smaller yard specializing in one or two standard type ships with a fairly high throughput so that one covered building dock or berth was sufficient.

At this point it may be convenient to mention the advantages and disadvantages of building docks as opposed to building berths. Building docks can be of advantage in the building of large vessels where launching costs are high, and there is a possibility of structural damage owing to the large stresses imposed by a conventional launch. They also give good crane clearance for positioning units. The greatest disadvantage of the building dock is its high initial cost.

Many yard re-constructions have incorporated undercover construction facilities in the form of docks or slipways within building halls. Others have building halls at the head of the slipway with advanced transfer systems installed so that the hull can be extruded out of the hall onto the slipway for

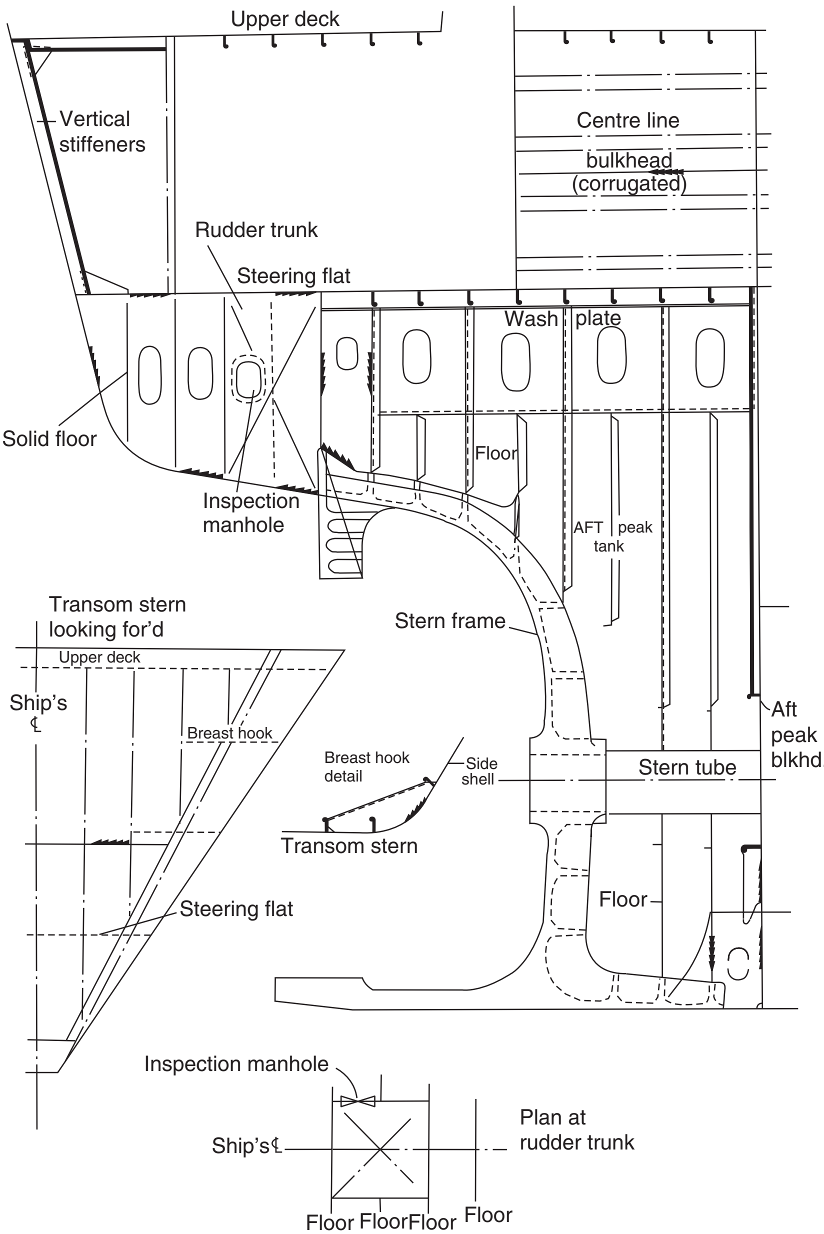


Figure 9.36 Transom stern.

launching. Such facilities permit ship construction in a factory type environment providing protection from the worst effects of weather and darkness.

9.4.4 Ship drawing office, Loftwork and CAD/CAM

This section describes the original functions of the ship drawing office and subsequent full or 10/1 scale lofting of the hull and its structural components and the current use by shipyards of computer aided design (CAD) for these purposes. The subsequent introduction and extensive use of computer aided manufacturing (CAM) in shipbuilding is also covered.

9.4.4.1 *Ship drawing office*

The ship drawing office was traditionally responsible for producing detailed working structural, general

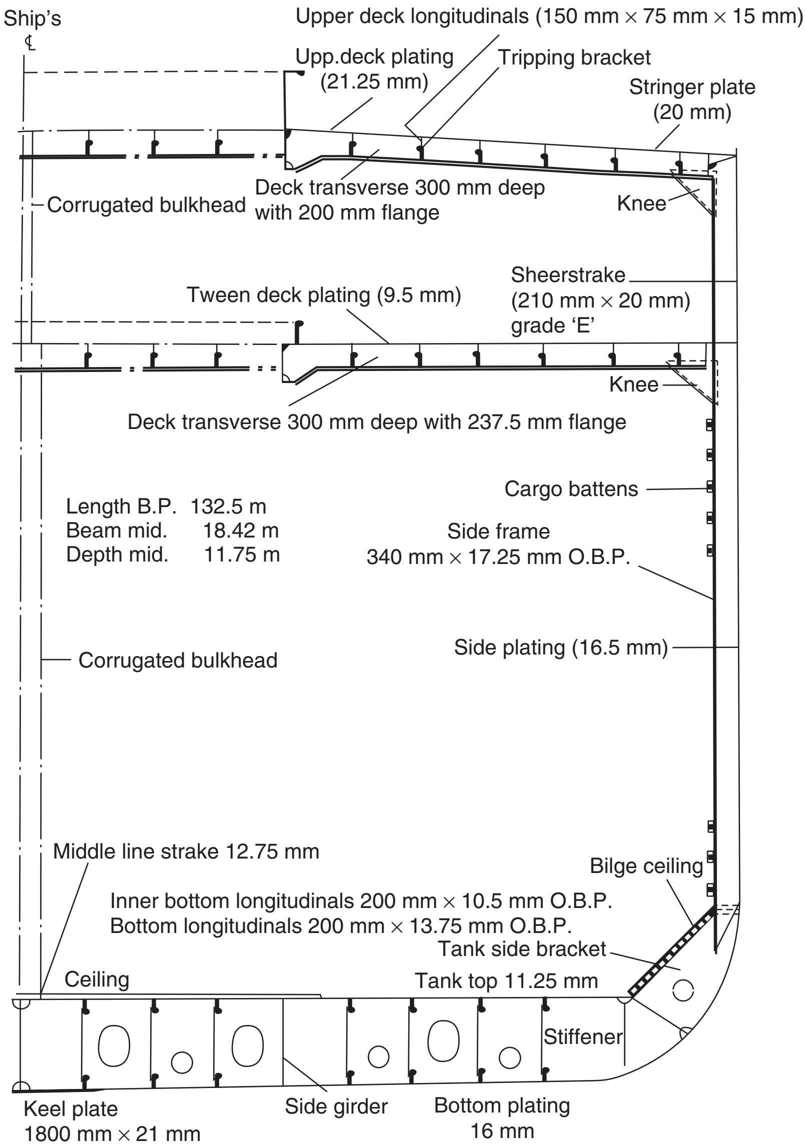
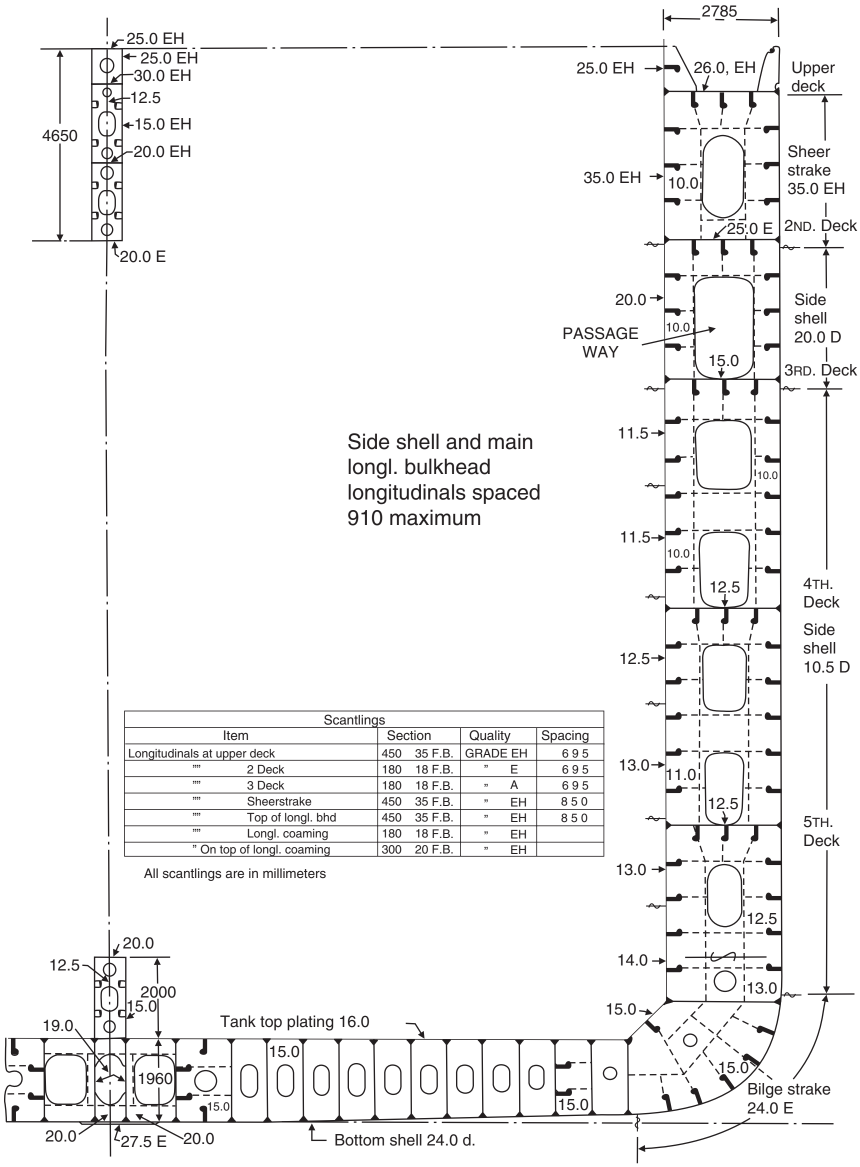


Figure 9.37 General cargo ship – midship section.

arrangement and outfit drawings for a new ship. It was also common practice for the drawing office to contain a material ordering department that would lift the necessary requirements from the drawings and progress them.

Structural drawings prepared by the drawing office would be in accordance with Lloyd's or other classification society rules and subject to their approval; also owner's additional requirements and standard shipyard practices would be incorporated in the drawings. General arrangements of all the accommodation and cargo spaces and stores would also be prepared, incorporating statutory requirements as well as any shipowner's requirements and standards. Outfit plans including piping arrangements, ventilation and air conditioning (which may be done by an outside contractor), rigging arrangements, furniture plans, etc. were also prepared. Two plans of particular significance were the ships 'lines plan' and 'shell expansion'.

Lines plan. A preliminary version of this was, in effect, prepared at the time of the conceptual design to give the required capacity, displacement and propulsive characteristics. It was subsequently



Scantlings			
Item	Section	Quality	Spacing
Longitudinals at upper deck	450 35 F.B.	GRADE EH	6 9 5
"" 2 Deck	180 18 F.B.	" E	6 9 5
"" 3 Deck	180 18 F.B.	" A	6 9 5
"" Sheerstrake	450 35 F.B.	" EH	8 5 0
"" Top of longl. bhd	450 35 F.B.	" EH	8 5 0
"" Longl. coaming	180 18 F.B.	" EH	
" On top of longl. coaming	300 20 F.B.	" EH	

All scantlings are in millimeters

Figure 9.38 Container ship – midship section.

refined during the preliminary design stage and following any tank testing or other method of assessing the hull's propulsive and seakeeping characteristics. The lines plan is a drawing, to a suitable scale, of the moulded lines of the vessel in plan, profile, and section. Transverse sections of the vessel at equally spaced stations between the after and forward perpendiculars are drawn to form what is known as the body plan. Usually ten equally spaced sections are selected with half ordinates at the ends where a greater change of shape occurs. A half transverse section only is drawn since the vessel is symmetrical about the centre line, and forward half sections are drawn to the right of the centre line with aft half sections to the left. Preliminary body plans are drawn initially to give the correct displacement, trim, capacity, etc., and must be laid off in plan and elevation to ensure fairness of the hull form. When the final faired body plan is available the full lines plan is completed showing also the profile or sheer plan of the vessel and the plan of the water-line shapes at different heights above the base.

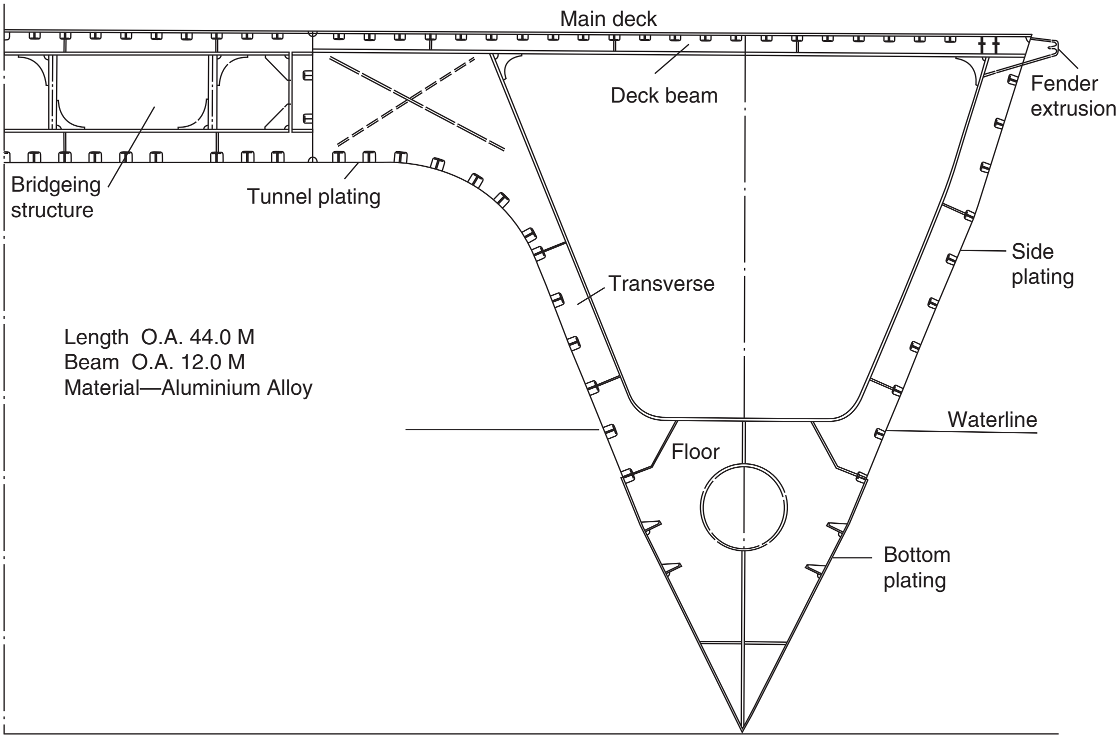


Figure 9.39 High-speed craft (catamaran) – section.

A lines plan is illustrated in Figure 9.44. The lines of the lateral sections in the sheer plan as indicated are referred to as 'bow lines' forward and 'buttock lines' aft. Bilge diagonals would be drawn with 'offsets' taken along the bilge diagonal to check fairness.

When the lines plan was completed manually the draughtsmen would compile a 'table of offsets', that is a list of half breadths, heights of decks and stringer, etc., at each of the drawn stations. These 'offsets' and the lines plan were then passed to loftsmen for full size or 10 to 1 scale fairing. Since the original lines plan was of necessity to a small scale which varied with the size of ship, the offsets tabulated from widely spaced stations and the fairing were not satisfactory for building purposes. The offsets used for building the ship would subsequently be lifted by the loftsman from the full size or 10 to 1 scale lines for each frame.

3-dimensional representation of shell plating. When preparing the layout and arrangement of the shell plating at the drawing stage it was often difficult to judge the line of seams and plate shapes with a conventional 2-dimensional drawing. Shipyards used to therefore make use of a 'half block model' which was in effect a scale model of half the ship's hull from the centre line outboard, mounted on a base board. The model was either made up of solid wooden sections with faired wood battens to form the exterior, or of laminated planes of wood faired as a whole. Finished with a white lacquer the model was used to draw on the frame lines, plate seams, and butts, lines of decks, stringers, girders, bulkheads, flats, stem and stern rabbets, openings in shell, bossings etc.

Shell expansion. The arrangement of the shell plating taken from a 3-dimensional model may be represented on a 2-dimensional drawing referred to as a shell expansion plan. All vertical dimensions in this drawing are taken around the girth of the vessel rather than their being a direct vertical projection. This technique illustrates both the side and bottom plating as a continuous whole. In Figure 9.45 a typical shell expansion for a tanker is illustrated. This also shows the numbering of plates, and lettering of plate strakes for reference purposes and illustrates the system where strakes 'run out' as the girth decreases forward and aft. This drawing was often subsequently retained by the shipowner to identify plates damaged in service. However a word of caution is necessary at this point because since prefabrication became the accepted practice any shell expansion drawing produced will generally have a numbering system related to the erection

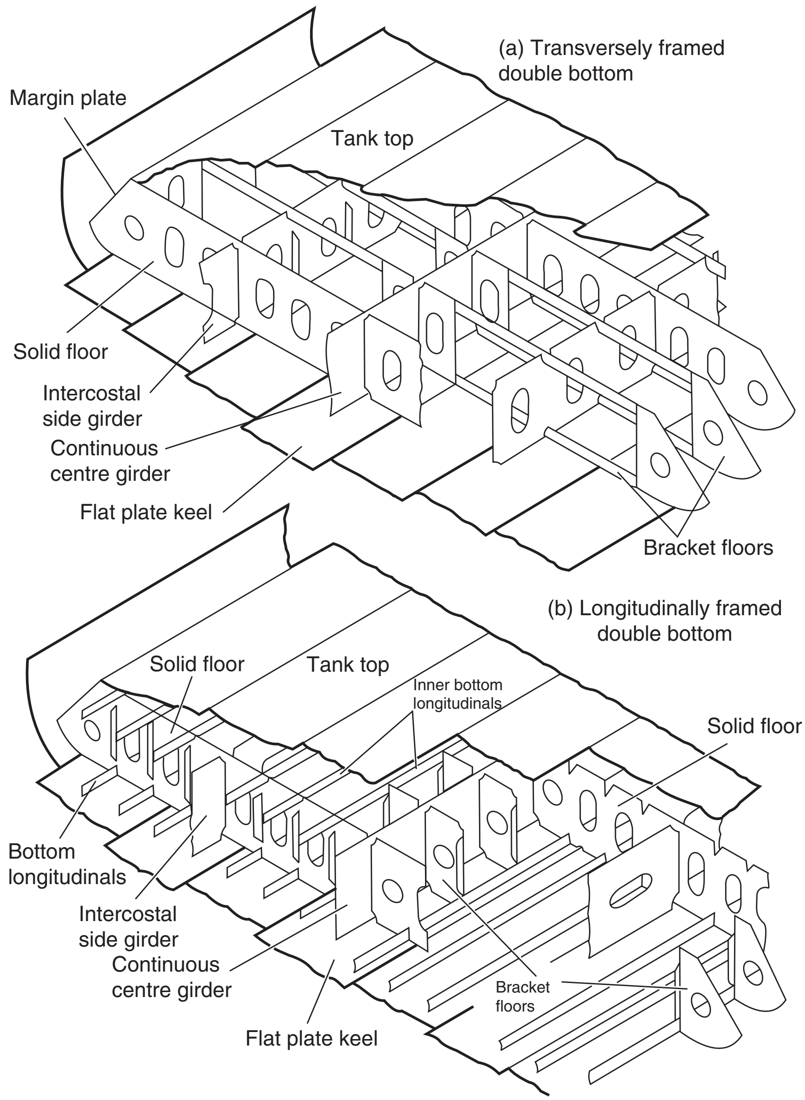


Figure 9.40 Double bottom construction.

of fabrication units rather than individual plates. However single plates were often marked in sequence to aid ordering and production identification.

9.4.4.2 *Loftwork following drawing office*

The mould loft in a shipyard was traditionally a large covered wooden floor area suitable for laying off ship details at full size.

When the loftsmen received the scale lines plan, and offsets from the drawing office, the lines would be laid off full size and faired. This would mean using a great length of floor even though a contracted sheer and plan were normally drawn, and aft and forward body lines were laid over one another. Body sections were laid out full size as they were faired to form what was known as a 'scrieve board'.

The scrieve board was used for preparing 'set bars' (curvature to match plate) and bevels (maintain web of frame perpendicular to ships centre line) for bending frames and for making templates and mouldings for plates which required cutting and shaping.

Shell plates were developed full size on the loft floor and wooden templates made so that these plates

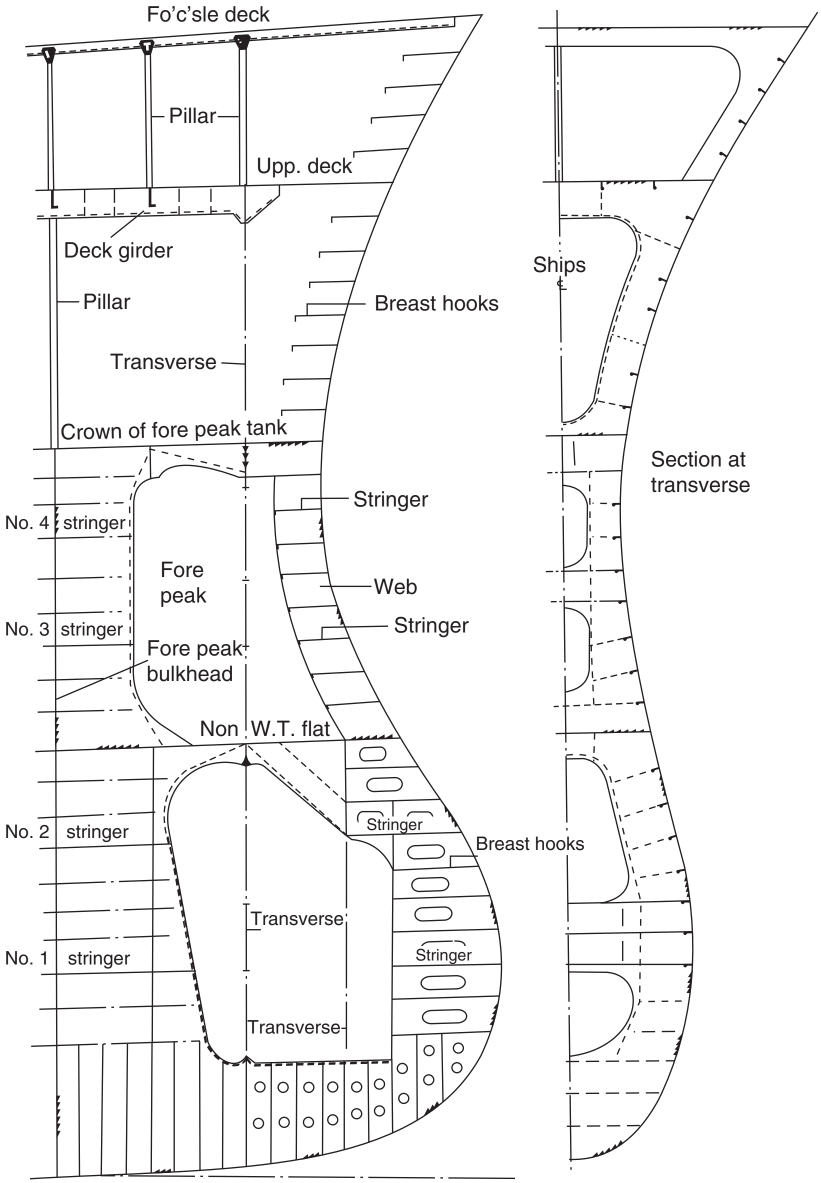


Figure 9.41 Bulbous bow.

could be marked and cut to the right shape before fitting to the framing on the berth.

10/1 scale lofting. In the late 1950s the 10/1 lofting system was introduced and was eventually widely adopted. This reduced the mould loft to a virtual drawing office and assisted in the introduction of production engineering methods. Lines could be faired on a 10/1 scale and a 10/1 scale scrieve board created. Many yards operated a flame profiling machine which used 10/1 template drawings to control the cutting operation. In preparing these template drawings the developed or regular shape of the plates was drawn in pencil on to special white paper or plywood sheet painted white, and then the outline was traced in ink on to a special transparent material. The material used was critical, having to remain constant in size under different temperature and humidity conditions and having a surface which would take ink without 'furring'. Many of the outlines of plates to be cut by the profiler could be traced directly from the scrieve board, for example floors and transverses.

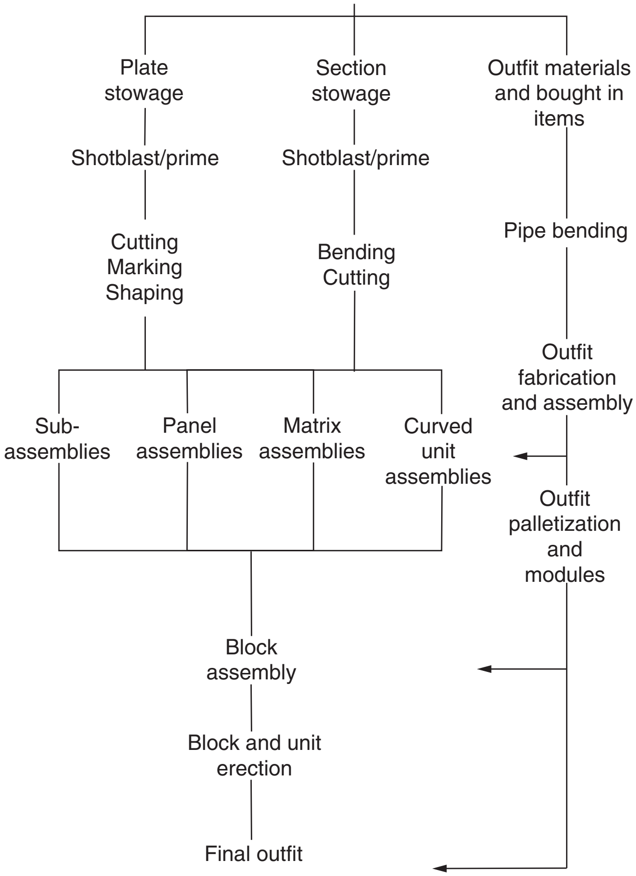


Figure 9.42 Shipbuilding process.

9.4.4.3 *Computer Aided Design (CAD)/Computer Aided Manufacturing (CAM)*

The first use of computers in the shipbuilding industry probably occurred in the 1960s and because of the high costs involved were only used by the largest shipbuilders running programs developed in-house on a mainframe or mini computer for hull lines fairing, hydrostatics, powering calculations etc. The hull design would have been drawn by hand and stored on the computer as tables of offsets.

In the late 1970s the graphics terminal and the Engineering Workstation became readily available and could be linked to a mini computer. These computers cost considerably less than the earlier mainframes and commercial ship design and construction software became available for them. The larger shipyards quickly adopted these systems. They developed further in the following two decades to run on UNIX Workstations and Windows NT machines and have expanded to cover virtually all the computing needs of a large shipyard.

The early 1980s saw the appearance of the Personal Computer (PC) and several low-cost software packages that performed simple hull design, hydrostatics and powering estimate tasks. These were popular with small shipyards and also reportedly with some larger shipyards for preliminary design work. They were however somewhat limited and incompatible so that it was difficult to build a system that covered all the shipyards CAD/CAM requirements. During the 1990s the available PC software standardized on hardware, operating systems, programming languages, data interchange file formats and hull geometry and are now widely used by naval architects and the ship and boat building industry in general.

Ship product model. Software systems for large shipbuilders is based on the concept of the 'Ship Product Model' in which the geometry and the attributes of all elements of the ship derived from the contract design and classification society structural requirements are stored. This model can be visualized at all stages and can be exploited to obtain information for production of the ship. See Figure 9.46.

At the heart of the 'Ship Product Model' is the conceptual creation of the hull form and its subsequent

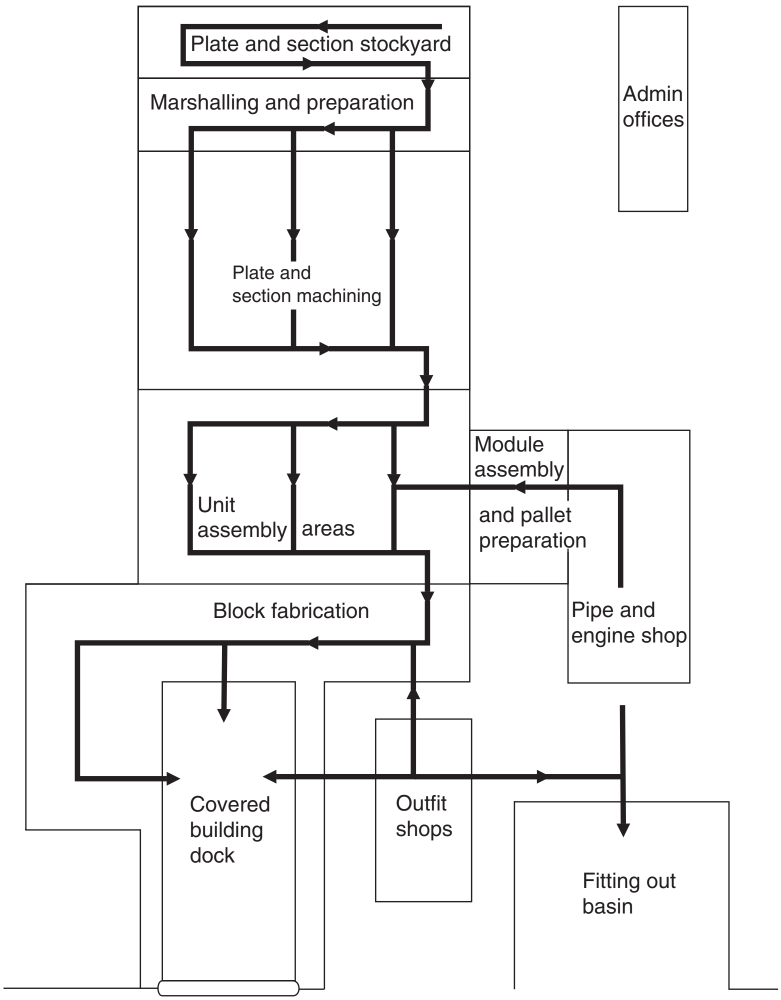


Figure 9.43 Shipyard layout.

fairing for production purposes which is accomplished without committing any plan to paper. This faired hull form is generally held in the computer system as a 'wire model' which typically defines the moulded lines of all structural items so that any structural section of the ship can be generated automatically from the 'wire model'. The model can be worked on interactively with other stored shipyard standards and practices to produce detailed arrangement and working drawings. The precision of the structural drawings generated enables them to be used with greater confidence than was possible with manual drawings and the materials requisitioning information can be stored on the computer to be interfaced with the shipyards commercial systems for purchasing and material control. Sub-assembly, assembly and block drawings can be created in 2-dimensional and 3-dimensional form and a library of standard production sequences and production facilities can be called up so that the draughtsman can ensure that the structural design uses the shipyards resources efficiently and follows established and cost effective practices. Weld lengths and types, steel weights and detailed parts lists can be processed from the information on the drawing and passed to the production control systems. A 3-dimensional steel assembly can be rotated by the draughtsman on screen to assess the best orientation for maximum downhand welding.

The use of 3-dimensional drawings is particularly valuable in the area of outfit drawings where items like pipework and ventilation/air-conditioning trunking can be 'sighted' in the 3-dimensional mode and more accurately measured before being created in the 2-dimensional drawing.

Stored information can be accessed so that lofting functions such as preparing information for bending

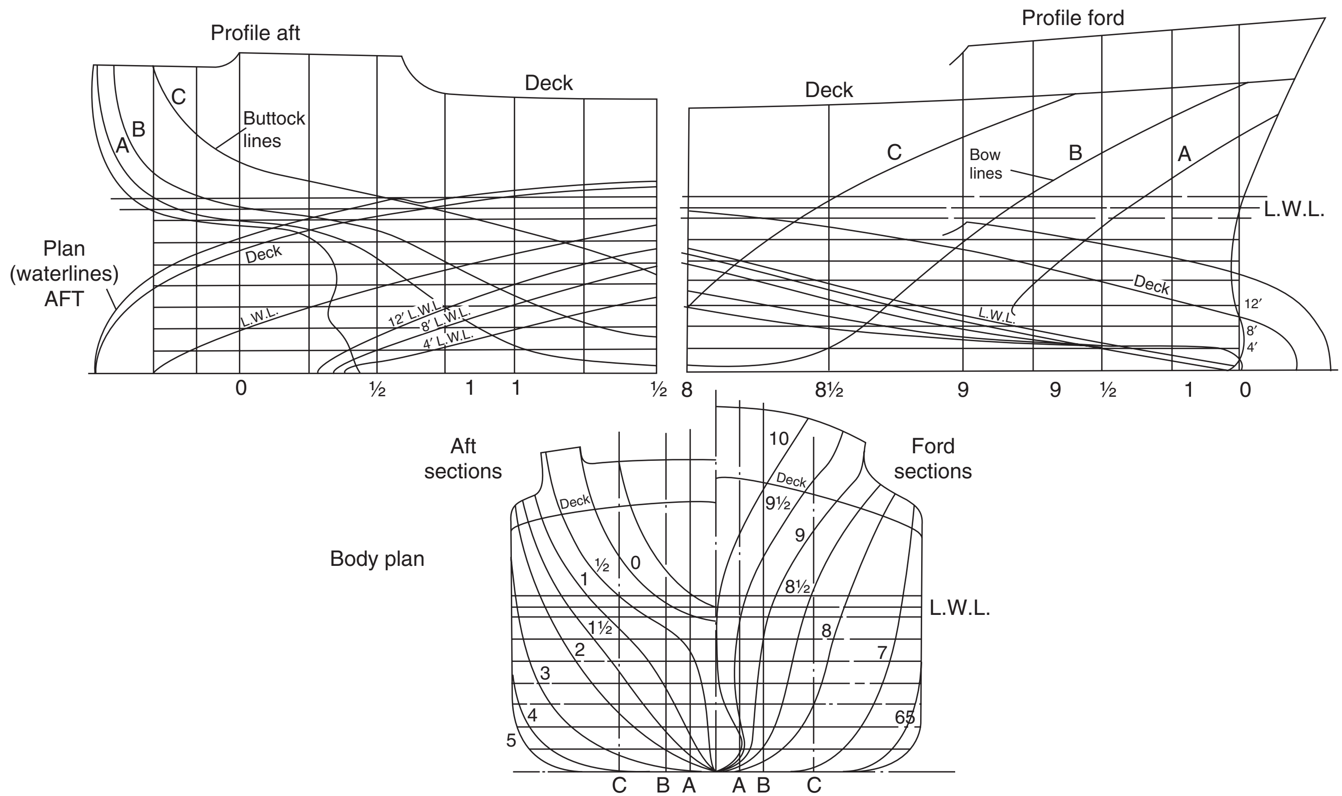
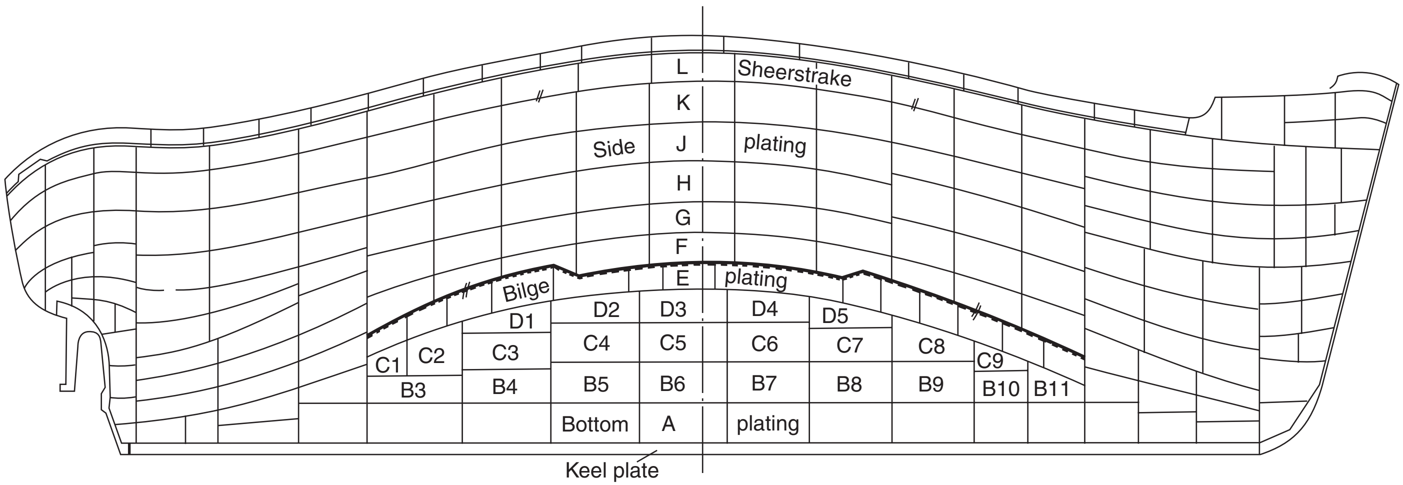


Figure 9.44 Lines plan.



Framing, stringers, decks and openings in side shell are also shown on the shell expansion but have been omitted for clarity

Figure 9.45 Shell expansion.

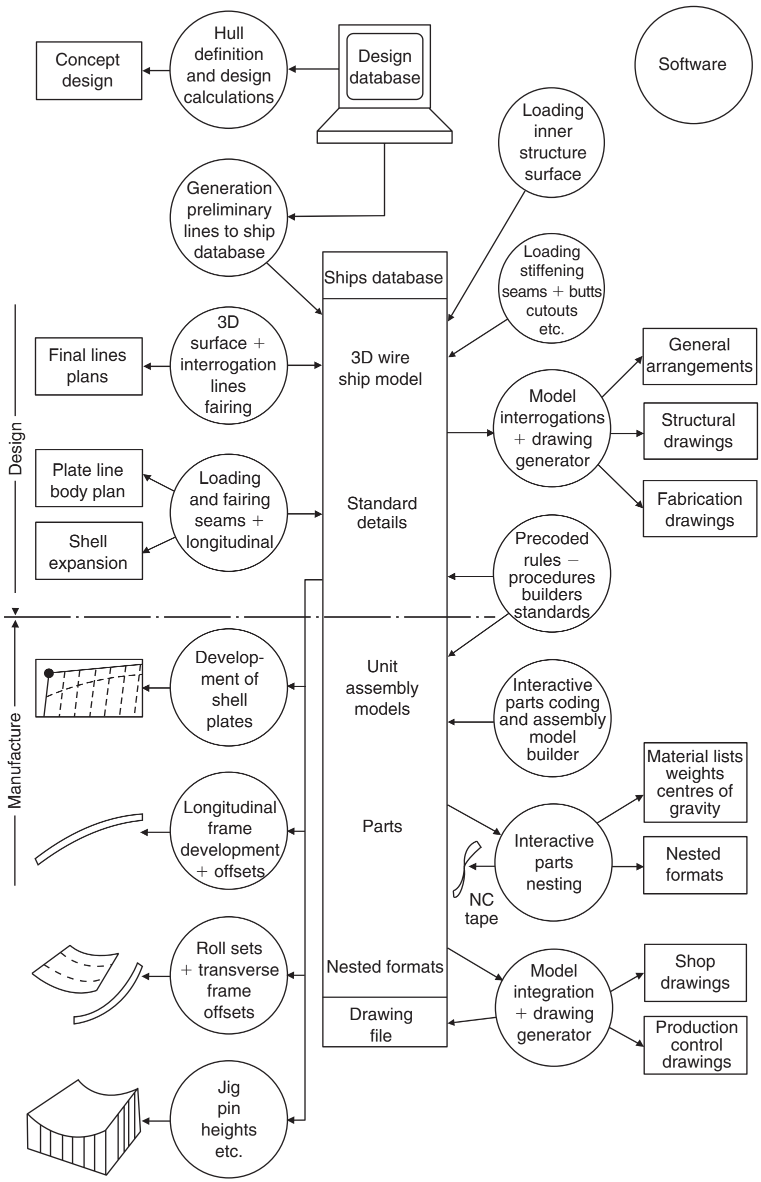


Figure 9.46 Integration of design and manufacture – ship product model.

frames and longitudinals, developing shell plates, and providing shell frame sets and rolling lines or heat line bending information for plates can be done via the interactive visual display unit.

For a numerically controlled profiling machine the piece parts to be cut can be 'nested', i.e. fitted into the most economic plate which can be handled by the machine with minimum wastage (see Figure 9.47). This can be done at the drawing stage when individual piece parts are abstracted for steel requisitioning and stored later being brought back to the screen for interactive nesting. The order in which parts are to be marked and cut can be defined by drawing the tool head around the parts on the graphics screen. When the burning instructions are complete the cutting sequence may be replayed and checked for errors. A check of the NC data can be carried out with a plotter. Instructions for cutting flame planed plates and subsequently joining them into panel assemblies and pin heights of jigs for setting up curved shell plates for welding framing and other members to them at the assembly stage can also be determined (see Figure 9.46).

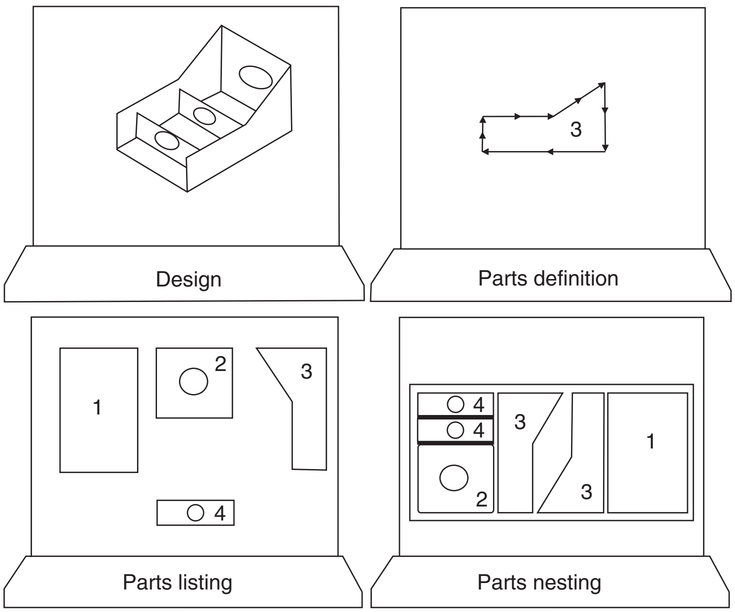


Figure 9.47 Assembly plate parts listing and nesting.

The basic Ship Product Model also contains software packages for the ships outfit including piping, electrical and heating, ventilating and air conditioning (HVAC) systems for a ship.

Further useful information on shipyard processes and production can be found in references such as Taggart (1980), Kuo *et al.* (1984), Torroja and Alonso (2000), Whitfield *et al.* (2003), Lamb *et al.* (2006) and Eyres (2007).

9.5 Ship economics

The following section is taken from Watson (1998). Further information on ship operational economics may be obtained from references such as Benford (1963), Goss (1965), Gilfillian (1969), Buxton (1972), Fisher (1972), Carreyette (1978), Erichsen (1989), Stopford (1997), Karayannis and Molland (2003) and Cullinane (2005).

9.5.1 Shipowners and operators

The operational economics of a ship can be looked at in a number of different ways depending on the type of trade in which it is used and how it is employed.

9.5.1.1 *Types of trade*

Whilst there is an enormous diversity in the type and size of ships, all are generally employed in one of five principal ways, namely as liners, cruise ships, industrial carriers, service vessels or as tramps. The first four of these categories can be classed as owner-operated ships, whilst the last category consists mainly of ships let out on charter.

(i) Liners
To be designated as a liner, a vessel must ply on a regular advertised service; examples are container ships and ferries, see Sections 2.2.2 and 2.2.3. Because ships providing this sort of service sail on scheduled dates and, when passengers are carried, at scheduled times, departing whether the ships are fully loaded or not, the cost of running a service of this type can be high. Freight rates and ticket prices must be set to achieve a satisfactory return over a period of time against the anticipated demand.

(ii) Cruise ships
The first cruises were offered by passenger liner companies using their liners either in their normal country to country service or on special voyages. These cruises were usually arranged at a time of year when passenger numbers in their normal services were likely to be on the low side.

With the decline of passenger services caused by the growth of air travel, passenger liners ceased to be available for use in this way and purpose built cruise liners started to make their appearance. These are now becoming more like floating hotels or holiday camps and the cruise business is currently one of the fastest growing areas of shipping. See also Section 2.2.6.

Typically, cruise ships undertake trips of one or two weeks duration generally steaming at night and with arrangements made for passengers to go ashore and see the sights and enjoy a new locality each day.

Although each cruise is a scheduled service, the fact that cruise schedules and itineraries can be changed at relatively short notice gives these ships an operational flexibility which liner services do not have.

(iii) Industrial carriers

A number of large companies with a substantial shipping requirement either for the import of their raw materials or for the export of their finished products or both own a number of ships to cover at least a baseload part of their shipping requirement.

Typical examples of this are the tanker fleets owned by oil companies; ships specially designed to carry iron ore and/or coal owned by steelmakers; and ships designed to carry cars in bulk owned by major car manufacturers, see Sections 2.2.4 and 2.2.5.

The owners of these ships generally assume total responsibility for all aspects of cost when the vessel is employed in their own trade. The object of such an ownership is to minimize the costs of an overall industrial process, but the lack of flexibility which has often been a characteristic of such operators has sometimes been found to do the opposite and this type of shipowner has been diminishing in recent years.

The U.S. anti-pollution laws have had a severe impact on some of the major oil companies who now refuse to trade with their own vessels in U.S. waters because of the virtually unlimited liability that applies there and instead charter in from traditional shipowners.

(iv) Service vessels

Very few, if any, service vessels carry cargo, their function being to supply services to other vessels or installations at sea. Examples of service vessels are tugs, dredgers, navigational service vessels, offshore safety vessels, etc. These services may be paid for directly as in the case of tugs or indirectly through port dues or taxation in some other cases. But the owners of all these ships need to calculate ship operating expenses on an owner operator basis.

(v) Tramps

A ship can be said to be tramping when it is prepared to go wherever a suitable cargo is available. Tramp ships can be employed in various ways under different types of charter which are explained in 9.5.1.2. Most bulk carriers and oil tankers, together with many small container ships and coasters operate as tramps, making this the method of employment of the majority of ships.

9.5.1.2 *Methods of employment*

An owner will generally employ a ship in one of four ways, namely: in his own trade, in tramp trades as an operator, or in tramp trades by time chartering or bareboat chartering the ship to another party. The extent to which an owner bears the costs of operations under each of these situations is discussed in the following paragraphs and is illustrated in Figure 9.48 which is a slightly modified version of a figure originally given in Dr. Buxton's 1972 R.I.N.A. paper 'Engineering economics applied to ship design', Buxton (1972) – a paper which, along with Dr. Buxton's earlier B.S.R.A. report 'Engineering economics and ship design', contributed substantially to this section.

(i) Ships used by an owner in his own trade

The types of trade in which ships are used by owners in their own trade have been outlined in 9.5.1.1. When ships are used in this way, the owner will generally assume total responsibility for all aspects of cost incurred.

(ii) Ships used by an owner as operator

An owner operator can arrange for the employment of a ship in a number of different ways, viz:

(i) by taking on Contracts of Affreightment to move a large volume of cargo in regular shipments of a set size, based on a set rate per tonne moved;

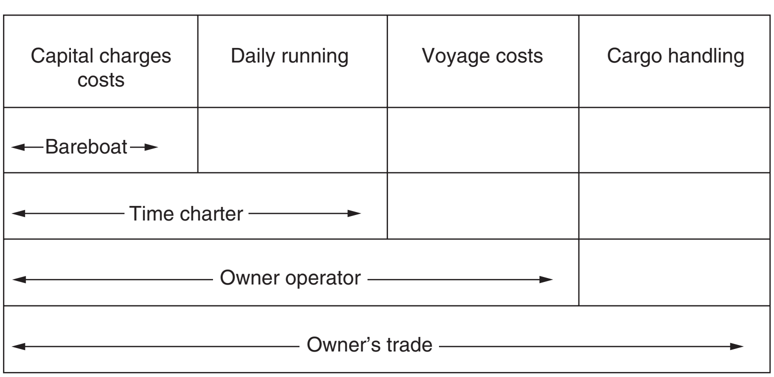


Figure 9.48 Changing responsibilities of the owner from bareboat to owner's trade.

(ii) by letting the ship on Voyage Charter to carry a single cargo on a set rate per tonne; or
(iii) by letting the ship for a single voyage on Time Charter for a set rate per day.

Under Contracts of Affreightment and Voyage Charters the owner will meet the capital cost, running costs and voyage costs (comprising port charges and bunkers). The terms of the charter will determine who pays the cargo handling costs as follows:

Gross terms (Gross)	Shipowner pays for loading and discharge
Free on board (FOB)	Charterer pays for loading
Free discharge (FD)	Charterer pays for discharge
Free in and out (FIO)	Charterer pays for loading and discharge

Under a single voyage time charter the charterer will meet the voyage costs as well as the cargo handling costs.

(iii) Tramping – let out on time charter
In a time charter, the shipowner undertakes to provide a ship for the charterer to use either for a fixed time of anything from a few months to 20 years or for a single round voyage.

The charterer is responsible for arranging cargoes and voyages during the charter and also for paying all voyage expenses including fuel, port and canal dues, cargo handling charges.

The shipowner provides the ship and crew and is responsible for the capital charges and daily running costs. Hire is only payable for time in service and ceases during breakdown and repair, although it continues if the ship is delayed in port or sails empty for reasons not attributable to the ship.

(iv) Tramping – let out on bareboat charter
In this case the charterer provides the crew and is responsible for maintenance with the shipowner's sole responsibility being the provision of the ship and meeting the capital charges. In effect the charterer uses the ship as if he owned it.

9.5.2 Economic criteria

9.5.2.1 *The basis of these criteria*

There are a number of different economic criteria which may be used to assess the likely success of a shipping investment or to compare the profitability of alternatives. These criteria should take account of:

- the time value of money,
- the full life of the investment,
- changes in items of income and expenditure which can be expected over the life,
- the economic facts of life such as interest rates; taxes; loans and investment grants.

The time value of money represents the fact that a sum of money available now is of much more value than the same sum not available for a number of years.

Interest is fundamental to the calculations whether there is a need to borrow or not. This takes account of the fact that if available cash is used the interest it might have earned is being foregone.

9.5.2.2 *Interest*

This may be simple or compound and the following relationships apply:

- Simple interest
  Total repayment after N years: $F = P\,(1 + N \cdot i)$
- Compound interest
  Total repayment after N years: $F = P\,(1 + i)^N$
  In this case the factor $(1 + i)^N$ is called the compound amount factor (CA), and P = original investment.

9.5.2.3 *Present worth*

The reciprocal of CA is called the present worth (PW) factor.

$$PW = 1/(CA) = (1 + i)^{-N}$$
$$P = (PW)F$$

The present worth of F, which includes all the accumulated interest is the same as the present sum of money P.

9.5.2.4 *Repayment of principal*

If the loan is repaid by annual instalments of principal plus interest, this may take two forms:

(i) principal repaid in equal instalments with interest being paid on the reducing balance; or
(ii) equal annual payments with interest predominating in the early years and capital repayments in the later years.

The concept of equal annual payments enables a present sum of money to be converted into an annual repayment sum spread over a number of years with the annual sum A being linked to the sum invested – the 'present sum P' by the capital recovery factor (CRF)

$$A = (CRF)P;\ \text{and}\ CRF = \frac{i(1+i)^N}{(1+i)^N - (1)}$$
$$\text{or}\ \frac{i}{1-(1+i)^{-N}}$$

The reciprocal of (*CRF*) is Series Present Worth factor (*SPW*). This is the multiplier required to convert a number of regular annual payments into a present sum.

9.5.2.5 *Sinking fund factor*

To find the annual sum (*A*) which accumulates to provide a future sum (*F*), this is multiplied by the sinking fund factor (*SF*)

$$A = F(SF); \text{ and } (SF) = \frac{i}{(1+i)^N - (1)}$$

The reciprocal of (*SF*) is the series compound amount factor (*SCA*)

$$SCA = 1/SF \text{ and } F = (SCA)A$$

With this brief introduction to, or refresher on, economics, the economic criteria commonly used in shipping can now be introduced.

9.5.2.6 *Net present value*

In this type of calculation the net present values (*NPV*) of income and expenditure are calculated over the assumed life of the ship (*N*) years. The final sum should be positive for the investment to be profitable at the assumed discount rate – or where alternatives are being compared it should be the larger sum.

$$NPV = \sum_{1}^{N} [PW \text{ (cargo tonnage } \times \text{ freight rate)} - PW\text{(operating costs)} - PW \text{ (ship acquisition costs)}]$$

9.5.2.7 *Required freight rate*

The required freight rate (*RFR*) is that which will produce a zero *NPV*, i.e. the break-even rate. Transposing the equation above gives:

$$RFR = \sum_{1}^{N} \left[\frac{(PW\text{(Operating costs)} + PW \text{ (Ship acquisition costs)}}{\text{Cargo tonnage}} \right]$$

9.5.2.8 *Yield*

In the above calculations a rate of interest must be assumed. If the freight rate is known or at least assumed, the rate at which money can be borrowed with $NPV = 0$, can be made the criterion.

9.5.2.9 *Inflation and exchange rates*

It is perhaps worth pointing out that economic forecasts of the sort described in the foregoing paragraphs are made on fixed money values. Inflation and the consequent reduction in the future value of money together with changes in exchange rates do not enter into these calculations although both of these must be estimated and taken into account in more detailed projections. This might be when fixing rates which are intended to apply over more than a limited period of time and/or when payments are to be made in a currency other than that in which the costs are incurred.

9.5.3 Operating costs

The next three sections as well as describing the components of operating costs try to suggest some ways of minimising these.

9.5.3.1 *Capital charges*

As Figure 9.48 shows, capital charges are included in the costing of all the different modes of ship operation and are in fact the only cost component in Bareboat chartering. Included in capital charges are:

– loan repayment
– loan interest
– profit
– taxes

9.5.3.2 *Capital amortization*

Loan interest and loan repayment can conveniently be taken together as capital amortisation.

The biggest component of capital charges is the repayment of the loan used to pay the shipbuilder. Payments to shipbuilders are almost invariably made in a number of instalments during the building period with a final instalment at the end of the guarantee period (usually a year after delivery).

Before the ship starts earning, its total cost will have increased above the tender price due to the interest payments on the sums paid out together with such other costs as those incurred in supervising construction, engaging the crew and in providing owners' supply items and initial stores.

Moreover, it will be an exceptional contract that does not result in some additional payments for changes in specification during building.

One obvious way to minimize capital charges is to keep the capital cost low, which may be achieved by good buying in relation to shipbuilding prices.

The initial building cost can, in principle, be kept down by building to a lower standard, although if this involves accepting that the ship will have a shorter than normal life this may not be a cost effective thing to do.

When considering capital economy measures, care must be taken to ensure that any lower standards adopted do not lead to higher operating costs that will negate any savings made.

The second largest component of capital charges is the sum paid in interest on the money borrowed to meet the costs incurred in building the ship and getting it into service.

Consequently, another way – and probably in the long term one of the most important ways – of minimizing capital charges, is by obtaining the most advantageous interest rates available.

Finally, at the end of whatever operating life is being assumed in the financial costing, the ship will still have a value, even if this is only as scrap, and an allowance for this should be made when assessing the cost of capital amortization.

The general assumption made in most financial assessments is that ships will have an operating life of 20 years. Although many continue in service for much longer periods, others become obsolete much earlier either as a result of changes in technology and/or in trading patterns and a 20 year period is probably a reasonable compromise.

9.5.3.3 *Profit and taxes*

The profit which the shipowner plans to make together with the taxes which this profit will incur forms the second part of capital charges.

9.5.3.4 *Depreciation*

Although depreciation does not enter into operating cost calculations, it seems desirable to include a short paragraph on the subject at this point as it does have a very significant effect on shipping company accounts, the tax paid and the profit made in particular years.

Depreciation is the process of writing off capital costs in company accounts. There are two classical methods of treating depreciation, namely:

(i) Straight line depreciation. If a 20-year life is assumed, the depreciation would be 5% per annum.
(ii) Declining balance depreciation. If a 15% per annum basis is assumed, then:

$$\begin{aligned}
&\text{Year 1: } 15\% \times 100 &&= 15\% \\
&\text{Year 2: } 15\% \times (100-15) &&= 12.75\% \\
&\text{Year 3: } 15\% \times (100-15-12.75) &&= 10.84\% \\
&\text{Year 10: } 3.52\% \\
&\text{Year 20: } 0.94\%
\end{aligned}$$

In most countries there are special provisions for the treatment of shipping depreciation from a taxation point of view. These treatments vary from country to country as do the rates of tax imposed.

Most of these treatments permit the writing off of a ship's capital cost at rather faster rates than the classical treatments. In general it pays a shipowner to depreciate as fast as the profits permit thus reducing or at least deferring tax payments.

9.5.3.5 *Ship values*

Although the book value of a ship at any time will be its original cost plus the cost of any repairs or alterations and minus the accumulated depreciation, the value of a ship as measured by its possible selling price is likely to fluctuate dramatically during its lifetime. This does not enter into operating cost calculations, although some owners significantly improve their profits by playing the market in this way.

9.5.4 Daily running costs

Included in daily running costs are:

– crew costs
– provisions and stores
– maintenance and repairs
– insurance
– administration and general charges

These costs are added for time charter calculations and of course also apply to voyage charters and owner operation. These are costs incurred whether the ship is at sea or in port.

9.5.4.1 *Crew costs*

The two major factors which determine crew costs today are crew numbers and the nationality of different sections of the officers and crew.

The effect of numbers is offset to some extent by the fact that a smaller crew will generally tend to have more 'chiefs' and fewer 'indians' and the fact that all the members of a reduced crew will (or certainly ought to) have a higher standard of training and as a consequence will (or ought to) be paid more *per capita.*

The automation and higher quality materials required to reduce watch-keeping and maintenance and thus enable the reduced crew to work the ship satisfactorily will increase the capital cost, whilst there is also likely to be a demand for higher class accommodation although this will be offset by the reduced number of cabins required.

9.5.4.2 *Provisions and stores*

Provisions are usually bought locally at the ship's trading ports and the annual cost is calculated on a per person per day basis.

Ships consume an extraordinary variety and quite considerable quantity of miscellaneous stores with the three most important items being chandlery, paint, chemicals and gases but with smaller sums being expended on such items as fresh water, laundry and charts.

Lubricating oil is sometimes included with this item, but it seems more logical to include it with bunkers.

9.5.4.3 *Maintenance and repair*

With today's small crews, maintenance at sea is necessarily limited, but careful planning by the ship's staff whilst at sea can greatly speed work carried out when in port and minimize its cost.

A big item under this heading is drydocking, but this is no longer an annual event with three or even five year intervals becoming usual.

Budgets for maintenance will generally include sums for work on the hull and superstructure, cargo spaces and systems, the main and auxiliary machinery, the electrical installation and the safety equipment plus survey fees.

Also included under this heading is the cost of riding squads which are now used to carry out maintenance and repairs which would have formerly been done by the crew but which is beyond the capability of the reduced crews of today.

9.5.4.4 *Insurance*

Insurance can be subdivided into Hull and P & I. The cost of Hull insurance is directly related to the capital cost of the ship with the insurance history of the managing company exercising a secondary effect. Costs have escalated significantly in recent years due to the number of major casualties and a generally ageing tonnage. Policies now provide for more deductibles and in the event of a claim these can increase running costs considerably.

P & I premiums have also increased greatly because of the U.S. Oil Pollution act and worries about crew standards.

9.5.4.5 *Administration and general charges*

Administration costs are a contribution to the office expenses of a shipping company or the fees payable to a management company plus a not inconsiderable sum for communications and sundries, together with flag charges.

Amongst the items included in general charges can be the cost of hiring items of ship's equipment such as the radio installation which are sometimes hired rather than bought as part of the ship.

The charge for the hire can be reduced by making a bulk deal for several ships with one company. The decision between buying and hiring demands reconsideration from time to time as prices, interest rates and tax measures change. At present the use of hired equipment is reducing.

It is also wise to allow in this heading a sum for exceptional items when preparing a cost estimate as regrettably only too often there will be something which cannot be foreseen.

9.5.5 Voyage costs

Included in voyage costs are:

– bunkers
– port and canal dues
– tugs, pilotage
– miscellaneous port expenses

These items are added when moving from a time charter to a voyage charter calculation and of course apply to owner operation.

9.5.5.1 *Bunkers*

(i) Oil fuel

The factors affecting oil fuel costs are the distance travelled, the average power used, the specific fuel consumption and the cost per tonne of fuel. The first of these can be minimized by good navigation which must also take into account favourable and adverse currents.

The second can be minimized by steaming at as slow a speed as enables the required schedule to be kept; by keeping the hull finish to a high standard of smoothness (a task that is much easier than it used to be with the latest long life and self polishing anti-fouling paints); and at an earlier stage, by good design of the ship's lines and the propeller.

Specific fuel consumption can be minimized at the design stage by a good choice of machinery and at the operating stage by keeping the engine well maintained.

The cost of fuel can be minimized by a careful choice of bunkering port, although any cost saving thus obtained must first meet any additional costs if a diversion is required or there is any reduction in cargo carrying capacity or increase in average voyage displacement increasing power and consumption. The fuel cost can also be reduced by the use of a poorer quality of fuel, although any saving must be assessed against any extra costs for purifiers, etc. needed for the fuel to be used and any increases in maintenance and repair costs that may result from its use. Bulk buying is yet another way of getting fuel at an advantageous price.

(ii) Diesel oil
Here the factors involved are the number of days, as generators are kept running in port as well as at sea, and the average electrical load. Because the cost of diesel oil is much higher than that of oil fuel it is advantageous to meet as much as possible of the electrical load by the use of shaft driven alternators.

(iii) Lubricating oil
Although the quantity of lubricating oil consumed is relatively small its high, unit cost results in it being a considerable item of expenditure. This item is sometimes included with stores, but as the usage depends on the distance travelled it seems better grouped with bunkers.

9.5.5.2 *Port and canal dues, pilotage, towage etc.*

(a) Port and canal dues
Port and canal dues depend on the tonnage of the vessel and on the trading pattern. Low gross and/or net tonnages are particularly important on some routes, such as those using the Suez or Panama canals or The St. Lawrence Seaway.

Booklets giving canal dues can be obtained from:

– Panama Canal Commission, Balboa, Republic of Panama (Fax: 507-272-2122)
– Suez Canal Authority, Ismailia, Arab Republic of Egypt (Fax: 064-320-784)
– St. Lawrence Seaway Authority, 360 Albert St, Ottawa, Canada (Fax: 613-598-4620)

(b) Pilotage costs
Pilotage costs are usually also assessed on gross tonnage but can be reduced in certain trades by having a ship's officer with a pilotage certificate where this procedure is followed.

(c) Towage and mooring costs
Tug charges can be eliminated or reduced if the ship is fitted with a bow thruster or approved high performance steering equipment.

The time spent in mooring can be reduced by fitting special deck machinery such as self-tensioning winches.

9.5.6 Cargo handling costs

Cargo handling costs include the costs arising from both loading and unloading cargo together with any claims that may arise relating to the cargo.

Cargo handling costs are excluded from voyage charter costs but have to be met in owner operation.

Cargo handling time can be reduced and with it the costs of this operation, by the provision of good cargo handling features such as:

1. large hatches giving good access;
2. shipside doors where appropriate;
3. hatch covers which can be speedily opened and closed;
4. fork lift trucks to speed stowage;
5. cargo handling cranes or derricks on the ship with a lift capacity optimized to the cargo carried and a speedy cycle time;
6. in appropriate cases by providing the ships with self discharging facilities.

Where the trade is based on a small number of specific ports there is the alternative of minimizing the ship cost and using shoreside cargo handling facilities.

Containerization or palletization of the cargo can make a step change in cargo handling time and cost.

9.6 Optimization in design and operation

9.6.1 Overview

Most design problems may be formulated as follows: determine a set of design variables (e.g. number of ships, individual ship size and speed in fleet optimization; main dimensions and interior subdivision of ship; scantlings of a construction; characteristic values of pipes and pumps in a pipe net) subject to certain relations between and restrictions of these variables (e.g. by physical, technical, legal, economical laws). If more than one combination of design variables satisfies all these conditions, we would like to determine that combination of design variables which optimizes some measure of merit (e.g. weight, cost, or yield).

9.6.2 Introduction to methodology of optimization

Optimization means finding the best solution from a limited or unlimited number of choices. Even if the number of choices is finite, it is often so large that it is impossible to evaluate each possible solution and then determine the best choice. There are, in principle, two methods of approaching optimization problems:

1. Direct search approach
   Solutions are generated by varying parameters either systematically in certain steps or randomly. The best of these solutions is then taken as the estimated optimum. Systematic variation soon

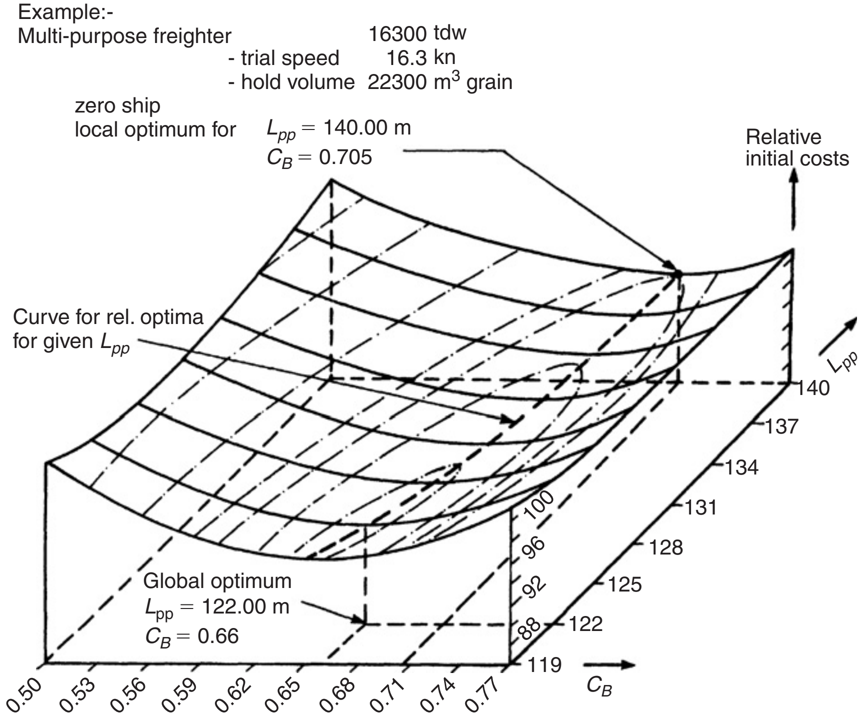


Figure 9.49 Example of overall costs dependent on length and block coefficient.

becomes prohibitively time consuming as the number of varied variables increases. Random searches are then employed, but these are still inefficient for problems with many design variables.

2. Steepness approach
   The solutions are generated using some information on the local steepness (in various directions) of the function to be optimized. When the steepness in all directions is (nearly) zero, the estimate for the optimum is found. This approach is more efficient in many cases. However, if several local optima exist, the method will 'get stuck' at the nearest local optimum instead of finding the global optimum, i.e. the best of *all* possible solutions. Discontinuities (steps) are problematic; even functions that vary steeply in one direction, but very little in another direction make this approach slow and often unreliable.

Most optimization methods in ship design are based on steepness approaches because they are so efficient for smooth functions. As an example consider the cost function varied over length L and block coefficient C_B (Figure 9.49). A steepness approach method will find quickly the lowest point on the cost function, if the function $K = f(C_B, L)$ has only one minimum. This is often the case.

Repeating the optimization with various starting points may circumvent the problem of 'getting stuck' at local optima. One option is to combine both approaches with a quick direct search using a few points to determine the starting point of the steepness approach. Also repeatedly alternating both methods – with the direct approach using a smaller grid scale and range of variation each time – has been proposed.

A pragmatic approach to treating discontinuities (steps) assumes first a continuous function, then repeats the optimization with lower and upper next values as fixed constraints and taking the better of the two optima thus obtained. Although, in theory, cases can be constructed where such a procedure will not give the overall optimum, in practice this procedure apparently works well.

The target of optimization is the objective function or criterion of the optimization. It is subject to boundary conditions or constraints. Constraints may be formulated as equations or inequalities. All technical and economical relationships to be considered in the optimization model must be known and expressed as functions. Some relationships will be exact, e.g. $\nabla = C_B \cdot L \cdot B \cdot T$; others will only be approximate, such as all empirical formulae, e.g. regarding resistance or weight estimates.

Procedures must be sufficiently precise, yet may not consume too much time or require highly detailed inputs. Ideally all variants should be evaluated with the same procedures. If a change of procedure is necessary, for example, because the area of validity is exceeded, the results of the two procedures must be correlated or blended if the approximated quantity is continuous in reality.

A problem often encountered in optimization is having to use unknown or uncertain values, e.g. future prices. Here plausible assumptions must be made. Where these assumptions are highly uncertain, it is common to optimize for several assumptions ('sensitivity study'). If a variation in certain input values only slightly affects the result, these may be assumed rather arbitrarily.

The main difficulty in most optimization problems does not lie in the mathematics or methods involved, i.e. whether a certain algorithm is more efficient or robust than others. The main difficulty lies in formulating the objective and all the constraints. If the human is not clear about his objective, the computer cannot perform the optimization. The designer has to decide first what he really wants. This is not easy for complex problems. Often the designer will list many objectives which a design shall achieve (e.g. see Section 9.2.1). This is then referred to in the literature as 'multi-criteria optimization', e.g. Sen (1992), Ray and Sha (1994). The expression is nonsense if taken literally. Optimization is only possible for one criterion, e.g. it is nonsense to ask for the best and cheapest solution. The best solution will not come cheaply, the cheapest solution will not be so good. There are two principle ways to handle 'multi-criteria' problems, both leading to one-criterion optimization:

1. One criterion is selected and the other criteria are formulated as constraints.
2. A weighted sum of all criteria forms the optimization objective. This abstract criterion can be interpreted as an 'optimum compromise'. However, the rather arbitrary choice of weight factors makes the optimization model obscure and we prefer the first option.

Throughout optimization, design requirements (constraints), e.g. cargo weight, deadweight, speed and hold capacity, must be satisfied. The starting point is called the 'basis design' or 'zero variant'. The optimization process generates alternatives or variants differing, for example, in main dimensions, form parameters, displacement, main propulsion power, tonnage, fuel consumption and initial costs. The constraints influence, usually, the result of the optimization. Figure 9.50 demonstrates, as an example, the effects of different optimization constraints on the sectional area curve.

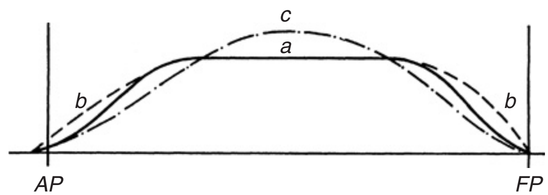


Figure 9.50 Changes produced in sectional area curve by various optimization constraints:
a is the basis form;
b is a fuller form with more displacement; optimization of carrying capacity with maximum main dimensions and variable displacement;
c is a finer form with the displacement of the basis form *a*, with variable main dimensions.

Optimized main dimensions often differ from the values found in built ships. There are several reasons for these discrepancies:

1. *Some built ships are suboptimal*
   The usual design process relies on statistics and comparisons with existing ships, rather than analytical approaches and formal optimization. Designs found this way satisfy the owner's requirements, but better solutions, both for the shipyard and the owner, may exist. Technological advances, changes in legislation and in economical factors (e.g. the price of fuel) are reflected immediately in an appropriate optimization model, but not when relying on partially outdated experience. Modern design approaches increasingly incorporate analyses in the design and compare more variants generated with the help of the computer. This should decrease the differences between optimization and built ships.
2. *The optimization model is insufficient*
   The optimization model may have neglected factors that are important in practice, but difficult to quantify in an optimization procedure, e.g. seakeeping behaviour, manoeuvrability, vibrational characteristics, easy cargo-handling. Even for directly incorporated quantities, often important relationships are overlooked, leading to wrong optima, e.g.:
   (a) A faster ship usually attracts more cargo, or can charge higher freight rates, but often income is assumed as speed independent.
   (b) A larger ship will generally have lower quay-to-quay transport costs per cargo unit, but time for cargo-handling in port may increase. Often, the time in port is assumed to be size independent.
   (c) In reefers the design of the refrigerated hold with regard to insulation and temperature requirements affects the optimum main dimensions. The additional investment and

annual costs have to be included in the model to obtain realistic results.

(d) The performance of a ship will often deteriorate over time. Operating costs will correspondingly increase, Malone *et al.* (1980), Townsin *et al.* (1981), but are usually assumed time independent.

The economic model may use an inappropriate objective function. Often there is confusion over the treatment of depreciation. This is not an item of expenditure, i.e. cash flow, but a book-keeping and tax calculation device, see Sections 9.5.3.4 and 9.6.4. The optimization model may also be based on too simplified technical relationships. Most of the practical difficulties boil down to obtaining realistic data to include in the analysis, rather than the mechanics of making the analysis. For example, the procedures for weight estimation, power prediction and building costs are quite inaccurate, which becomes obvious when the results of different published formulae are compared. The optimization process may now just maximize the error in the formulae rather than minimize the objective.

The result of the optimization model should be compared against built ships. Consistent differences may help to identify important factors so far neglected in the model. A sensitivity analysis concerning the underlying estimation formulae will give a bandwidth of 'optimal' solutions and any design within this bandwidth must be considered as equivalent. If the bandwidth is too large, the optimization is insignificant.

A critical view on the results of optimization is recommended. But properly used optimization may guide us to better designs than merely reciprocating traditional designs. The ship main dimensions should be appropriately selected by a naval architect who understands the relationships of various variables and the pitfalls of optimization. An automatic optimization does not absolve the designer of his responsibility. It only supports him in his decisions.

9.6.3 Scope of application in ship design

Formal optimization of the lines including the bulbous bow even for fixed main dimensions is beyond our current computational capabilities. Although such formal optimization has been attempted using CFD methods, the results were not convincing despite high computational effort, Janson (1997). Instead, we will focus here on ship design optimization problems involving only a few (less than 10) independent variables and rather simple functions. A typical application would be the optimization of the main dimensions. However, optimization may be applied to a wide variety of ship design problems ranging from fleet optimization to details of structural design.

In fleet optimization, the objective is often to find the optimum number of ships, ship speed and capacity without going into further details of main dimensions, etc. A ship's economic efficiency is usually improved by increasing its size, as specific cost (cost per unit load, e.g. per TEU or per ton of cargo) for initial cost, fuel, crew, etc., decrease. However, dimensional limitations restrict size. The draught (and thus indirectly the depth) is limited by channels and harbours. However, for draught restrictions one should keep in mind that a ship is not always fully loaded and harbours may be dredged to greater draughts during the ship's life. The width of tankers is limited by building and repair docks. The width of containerships is limited by the span of container bridges. Locks restrict all the dimensions of inland vessels. In addition, there are less obvious aspects limiting the optimum ship size:

1. The limited availability of cargo coupled to certain expectations concerning frequency of departure limits the size on certain routes.
2. Port time increases with size, reducing the number of voyages per year and thus the income.
3. The shipping company loses flexibility. Several small ships can service more frequently various routes/harbours and will thus usually attract more cargo. It is also easier to respond to seasonal fluctuations.
4. Port duties increase with tonnage. A large ship calling on many harbours may have to pay more port dues than several smaller ships servicing the same harbours in various routes, thus calling each in fewer harbours.
5. In container line shipping, the shipping companies offer door-to-door transport. The costs for feeder and hinterland traffic increase if large ships only service a few 'hub' harbours and distribute the cargo from there to the individual customer. Costs for cargo-handling and land transport then often exceed savings in shipping costs.

These considerations largely concern shipping companies in optimizing the ship size. Factors favouring larger ship size are, Buxton (1976):

- Increased annual flow of cargo.
- Faster cargo-handling.
- Cargo available one way only.
- Long-term availability of cargo.
- Longer voyage distance.
- Reduced cargo-handling and stock-piling costs.
- Anticipated port improvements.
- Reduced unit costs of building ships.
- Reduced frequency of service.

We refer to Benford (1963, 1965) for more details on selecting ship size.

After the optimum size, speed, and number of ships has been determined along with some other specifications, the design engineer at the shipyard is usually tasked to perform an optimization of the main dimensions as a start of the design. Further stages of the design will involve local hull shape, e.g. design of the bulbous bow lines, structural design, etc. Optimization of structural details often involves only a few variables and rather exact functions. Söding (1977) presents as an example the weight optimization of a corrugated bulkhead. Other examples are found in Liu *et al*. (1981) and Winkle and Baird (1985).

For the remainder of the Section we will discuss only the optimization of main dimensions for a single ship. Pioneering work in introducing optimization to ship conceptual design in Germany has been performed by the Technical University of Aachen (Schneekluth, 1957, 1967; Malzahn *et al.*, 1978). Such an optimization varies technical aspects and evaluates the result from an economic viewpoint. Fundamental equations (e.g. $\nabla = C_B \cdot L \cdot B \cdot T$), technical specifications/constraints, and equations describing the economical criteria form a more or less complicated system of coupled equations, which usually involve nonlinearities. Gudenschwager (1988) gives an extensive optimization model for Ro-Ro ships with 57 unknowns, 44 equations, and 34 constraints.

To establish such complicated design models, it is recommended to start with a few relations and design variables, and then to improve the model step by step, always comparing the results with the designer's experience and understanding the changes relative to the previous, simpler model. This is necessary in a complicated design model to avoid errors or inaccuracies which cannot be clarified or which may even remain unnoticed without applying this stepwise procedure. Design variables which involve step functions (number of propeller blades, power of installed engines, number of containers over the width of a ship, etc.) may then be determined at an early stage and can be kept constant in a more sophisticated model, thus reducing the complexity and computational effort. Weakly variation-dependent variables or variables of secondary importance (e.g. displacement, underdeck volume, stability) should only be introduced at a late stage of the development procedure. The most economic solution often lies at the border of the search space defined by constraints, e.g. the maximum permissible draught or Panamax width for large ships. If this is realized in the early cycles, the relevant variables should be set constant in the optimization model in further cycles. Keane *et al.* (1991) discuss solution strategies of optimization problems in more detail.

Simplifications can be retained if the associated error is sufficiently small. They can also be given subsequent consideration.

9.6.4 Economic basics for optimization

9.6.4.1 *Discounting*

An outline of the economic criteria has been given in Section 9.5.2. For purposes of optimization, all payments are discounted, i.e. converted by taking account of the interest, to the time when the vessel is commissioned. The rate of interest used in discounting is usually the market rate for long-term loans. Discounting decreases the value of future payments and increases the value of past payments. Individual payments thus discounted are, for example, instalments for the new building costs and the re-sale price or scrap value of the ship. The present value (discounted value) K_{pv} of an individual payment K paid N years later—e.g. scrap or re-sale value—is:

$$K_{pv} = K \cdot \frac{1}{(1+i)^N} = K \cdot \mathrm{PW}$$

where i is the interest rate. PW is the present worth factor. For an interest rate of 8%, the PWF is 0.2145 for an investment life of 20 years, and 0.9259 for 1 year. If the scrap value of a ship after 20 years is 5% of the initial cost, the discounted value is about 1%. Thus the error in neglecting it for simplification is relatively small.

A series of constant payments k is similarly discounted to present value K_{pv} by:

$$K_{pv} = k \cdot \frac{(1+i)^N \cdot i}{(1+i)^N - 1} = k \cdot \mathrm{CRF}$$

CRF is the capital recovery factor. The shorter the investment life, the greater is the CRF at the same rate of interest. For an interest rate of 8%, the CRF is 0.1018 for 20 years and 1.08 for 1 year of investment life.

The above formulae assume payment of interest at the end of each year. This is the norm in economic calculations. However, other payment cycles can easily be converted to this norm. For example, for quarterly payments divide i by 4 and multiply N by 4 in the above formulae.

For costs incurred at greater intervals than years, or on a highly irregular basis, e.g. large-scale repair work, an annual average is used. Where changes in costs are anticipated, future costs should be entered at the average annual level as expected. Evaluation of individual costs is based on present values which may be corrected if recognizable longer-term trends exist. Problems are:

1. The useful life of the ship can only be estimated.
2. During the useful life, costs can change with the result that cost components may change in absolute terms and in relation to each other. After the oil crisis of 1973, for example, fuel costs rose dramatically.

All expenditure and income in a ship's life can thus be discounted to a total 'net present value' (NPV). Only the cash flow (expenditure and income) should be considered, not costs which are used only for accounting purposes.

Yield is the interest rate *i* that gives zero NPV for a given cash flow. Yield is also called Discounted Cash Flow Rate of Return, or Internal Rate of Return. It allows comparisons between widely different alternatives differing also in capital invested. In principle, yield should be used as the economic criterion to evaluate various ship alternatives, just as it is used predominantly in business administration as the benchmark for investments of all kinds. The operating life should be identical for various investments then. Unfortunately, yield depends on uncertain quantities like future freight rates, future operating costs, and operating life of a ship. It also requires the highest computational effort as building costs, operating costs and income must all be estimated.

Other economic criteria which consider the time value of money include NPV, NPV/investment, or Required Freight Rate (= the freight rate that gives zero NPV); they are discussed in more detail by Buxton (1972, 1976). The literature is full of long and rather academic discussions on what is the best criterion. But the choice of the economic criterion is actually of secondary importance in view of the possible errors in the optimization model (such as overlooking important factors or using inaccurate relationships).

Discounting decreases the influence of future payments. The initial costs, not discounted, represent the single most important payment and are the least afflicted by uncertainty. (Strictly speaking, the individual instalments of the initial costs should be discounted, but these are due over the short building period of the ship.) The criterion 'initial costs' simplifies the optimization model, as several variation-independent quantities can be omitted. Initial costs have often been recommended as the best criterion for shipyard as this maximises the shipyard's profit. This is only true if the price for various alternatives is constant. However, in modern business practice the shipyard has to convince the shipowner of its design. Then price will be coupled to expected cash flow.

In summary, the criterion for optimization should usually be yield. For a simpler approach, which may often suffice or serve in developing the optimization model, initial costs may be minimized.

9.6.4.2 *Initial costs (building costs)*

Building costs can be roughly classified into:

- Direct labour costs.
- Direct material costs (including services bought).
- Overhead costs.

Overhead costs are related to individual ships by some appropriate key, for example equally among all ships built at the accounting period, proportional to direct costs, etc. See also Carreyette (1978) for a discussion of costs.

For optimization, the production costs are divided into (Figure 9.51):

1. *Variation-dependent costs*
   Costs which depend on the ship's form:
   (a) Cost of hull.
   (b) Cost of propulsion unit (main engine).
   (c) Other variation-dependent costs, e.g. hatchways, pipes, etc.
2. *Variation-independent costs*
   Costs which are the same for every variant, e.g. navigation equipment, living quarters, etc.

Buxton (1976) gives some simple empirical estimates for these costs.

Building costs are covered by own capital and loans. The source of the capital may be disregarded. Then also interest on loans need not be considered in the cash flow. The yield on the capital should then be larger than alternative forms of investment, especially the interest rate of long-term loans. This approach is too simple for an investment decision, but suffices for optimizing the main dimensions.

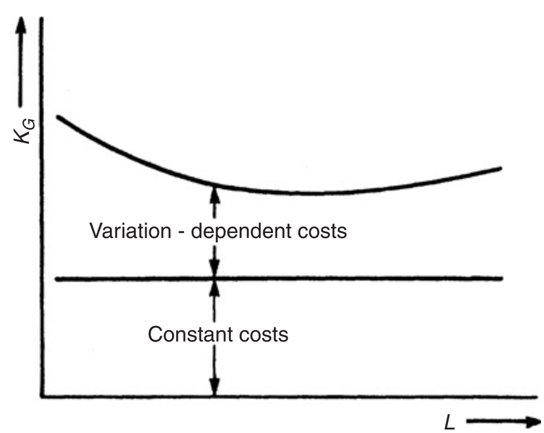


Figure 9.51 Division of costs into length-dependent and length-independent.

Typically 15–45% of the initial costs are attributable to the shipyard, the rest to outside suppliers. The tendency is towards increased outsourcing. Of the wages paid by the shipyard, typically 20% are allotted to design and 80% to production for one-of-a-kind cargo ships, while warships feature typically a 50:50 proportion.

Determining the variation-dependent costs
Superstructure and deckhouses are usually assumed to be variation-independent when considering variations of main dimensions. The variation-dependent costs are:

1. The hull steel costs.
2. The variation-dependent propulsion unit costs.
3. Those components of equipment and outfit which change with main dimensions.

The steel costs
The yards usually determine the costs of the processed steel in two separate groups:

1. The cost of the unprocessed rolled steel. The costs of plates and rolled sections are determined separately using prices per ton. The overall weight is determined by the steel weight calculation. The cost of wastage must be added to this.
2. Other costs. These comprise mainly wages. This cost group depends on the number of man-hours spent working on the ship within the yard. The numbers differ widely, depending on the production methods and complexity of construction. As a rough estimate, 25–35 man-hours/t for containerships are cited in older literature. There are around 30–40% more man-hours/t needed for constructing the superstructure and deckhouses than for the hull, and likewise for building the ship's ends as compared with the parallel middlebody. The amount of work related to steel weight is greater on smaller ships. For example, a ship with 70 000 m^3 underdeck volume needs 15% less manufacturing time per ton than a ship with 20 000 m^3, Kerlen (1985).

For optimization, it is more practical to form 'unit costs per ton of steel installed', and then multiply these unit costs by the steel weight. These unit costs can be estimated as the calculated production costs of the steel hull divided by the net steel weight. Kerlen (1985) gives the specific hull steel costs as:

$$k_{St}[\mathrm{MU/t}] = k_0 \cdot \left(\frac{4}{\sqrt[3]{L/m}} + \frac{3}{L/m} + 0.2082 \right) \cdot \left(\frac{3}{2.58 + C_B^2} - 0.07 \cdot \frac{0.65 - C_B}{0.65} \right)$$

k_0 represents the production costs of a ship 140 m in length with $C_B = 0.65$. The formula is applicable for ships with $0.5 \le C_B \le 0.8$ and $80\,\mathrm{m} \le L \le 200\,\mathrm{m}$. The formula may be modified, depending on the material costs and changes in work content.

Propulsion unit costs
For optimization of main dimensions, the costs of the propulsion plant may be assumed to vary continuously with propulsion power. They can then be obtained by multiplying propulsion power by unit costs per unit of power. A further possibility is to use the catalogue prices for engines, gears and other large plant components in the calculation and to take account of other parts of the machinery by multiplying by an empirical factor. Only those parts which are functions of the propulsion power should be considered. The electrical plant, counted as part of the engine plant in design – including the generators, ballast water pipes, valves and pumps – is largely variation-independent.

The costs of the weight group 'equipment and outfit'
Whether certain parts are so variation-dependent as to justify their being considered depends on the ship type. For optimization of initial costs, the equipment can be divided into three groups:

1. Totally variation-independent equipment, e.g. electronic units on board.
2. Marginally variation-dependent equipment, e.g. anchors, chains and hawsers which can change if in the variation the classification numeral changes. If variation-dependence is not pronounced, the equipment in question can be omitted.
3. Strongly variation-dependent equipment, e.g. the cost of hatchways rises roughly in proportion to the hatch length and the 1.6th power of the hatch width, i.e. broad hatchways are more expensive than long, narrow ones.

Relationship of unit costs
Unit costs relating to steel weight and machinery may change with time. However, if their ratio remains constant, the result of the calculation will remain unchanged. If, for example, a design calculation for future application assumes the same rates of increase compared with the present for all the costs entered in the calculation, the result will give the same main dimensions as a calculation using only current data.

9.6.4.3 *Annual income and expenditure*

The income of cargo ships depends on the amount of cargo and the freight rates. Both should be a function of speed in a free market. At least the interest of the tied-up capital cost of the cargo should be included as a lower estimate for this speed dependence. The issue will be discussed again in Section 9.6.5. for the effect of speed.

Expenditure over the lifetime of a ship includes:

1. *Risk costs*
   Risk costs relating to the ship consist mainly of the following insurance premiums:
   - Insurance on hull and associated equipment.
   - Insurance against loss or damage by the sea.
   - Third-party (indemnity) insurance.

   Annual risk costs are typically 0.5% of the production costs.
2. *Repair and maintenance costs*
   The repair and maintenance costs can be determined using operating cost statistics from suitable basis ships, usually available in shipping companies.
3. *Fuel and lubricating costs*
   These costs depend on engine output and operating time.
4. *Crew costs*
   Crew costs include wages and salaries including overtime, catering costs, and social contributions (health insurance, accident and pension insurance, company pensions). Crewing requirements depend on the engine power, but remain unchanged for a wide range of outputs for the same system. Thus crew costs are usually variation-independent. If the optimization result shows a different crewing requirement from the basis ship, crew cost differences can be included in the model and the calculation repeated.
5. *Overhead costs*
   - Port duties, lock duties, pilot charges, towage costs, haulage fees.
   - Overheads for shipping company and broker.
   - Hazard costs for cargo (e.g. insurance, typically 0.2–0.4% of cargo value).

   Port duties, lock duties, pilot charges and towage costs depend on the tonnage. The proportion of overheads and broker fees depend on turnover and state of employment. All overheads listed here are variation-independent for constant ship size.
6. *Costs of working stock and extra equipment*
   These costs depend on ship size, size of engine plant, number of crew, etc. The variation-dependence is difficult to calculate, but the costs are small in relation to other cost types mentioned. For this reason, differences in working-stock costs may be neglected.
7. *Cargo-handling costs*
   Cargo-handling costs are affected by ship type and the cargo-handling equipment both on board and on land. They are largely variation-independent for constant ship size.

Taxes, interest on loans covering the initial building costs and inflation have only negligible effects on the optimization of main dimensions and can be ignored.

9.6.4.4 *The 'cost-difference' method*

Cash flow and initial costs can be optimized by considering only the differences with respect to the 'basis ship'. This simplifies the calculation as only variation-dependent items remain. The difference costs often give more reliable figures.

Objective function for initial costs optimization
The initial difference costs consist of the sum of hull steel difference costs and propulsion unit difference costs:

$$\begin{aligned}\Delta K_G[\text{MU}] &= W_{St_0} \cdot k_{St_0} - W_{St_n} \cdot k_{St_n} + \Delta K_M \cdot C_M \\ &= W_{St_0} \cdot k_{St_0} - W_{St_n} \cdot k_{St_n} + \Delta P_B \cdot k_M \cdot C_M\end{aligned}$$

ΔK_G	[MU]	difference costs for the initial costs
W_{St_0}	[t]	hull steel weight for basis variant
W_{St_n}	[t]	hull steel weight for variant *n*
k_{St}	[MU/t]	specific costs of installed steel
ΔK_M	[MU]	difference costs for the main engine
C_M		factor accounting for the difference costs of the 'remaining parts' of the propulsion unit
ΔP_B	[kW]	difference in the required propulsion power
k_M	[MU/kW]	specific costs of engine power

In some cases the sum of the initial difference costs should be supplemented further by the equipment difference costs.

Objective function for yield optimization
The yield itself is not required, only the variant which maximizes yield. Again, only the variation-dependent cash flow needs to be considered. The most important items are the differences in:

1. Initial costs
2. Fuel and lubricant costs
3. Repair and insurance costs
4. Net income if variation-dependent

The power requirements are a function of trial speed, therefore the initial costs of the propulsion unit depend on the engine requirements under trial speed conditions. The fuel costs should be related to the service speed. The annual fuel and lubricant costs then become:

$$k_{f+l}[\text{MU/yr}] = P_{B,D} \cdot F \cdot (k_f \cdot s_f + k_l \cdot s_l)$$

$P_{B,D}$	[kW]	brake power at service speed
F	[h]	annual operating time
k_f	[MU/t]	cost of 1 t of fuel (or heavy oil)
s_f	[t/kWh]	specific fuel consumption
k_l	[MU/t]	cost of 1 t of lubricating oil
s_l	[t/kWh]	specific lubricant consumption

9.6.4.5 *Discontinuities in propulsion unit costs*

Standardised propulsion unit elements such as engines, gears, etc. introduce steps in the cost curves (Figures 9.52 and 9.53). The stepped curve can have a minimum on the faired section or at the lower point of a break. With the initial costs, the optimum is always situated at the beginning of the curve to the right of the break. Changing from a smaller to a larger engine reduces the engine loading and thus repair costs. The fuel costs are also stepped where the number of cylinders changes (Figure 9.54). At one side of the break point the smaller engine is largely fully loaded. On the other side, the engine with one more cylinder has a reduced loading, i.e. lower fuel consumption. Thus when both initial costs and annual costs are considered the discounted cash flow is quasi-continuous.

The assumption of constant speed when propulsion power is changed in steps is only an assumption for comparison when determining the optimum main dimensions. In practice, if the propulsion plant is not fully employed, a higher speed is adopted.

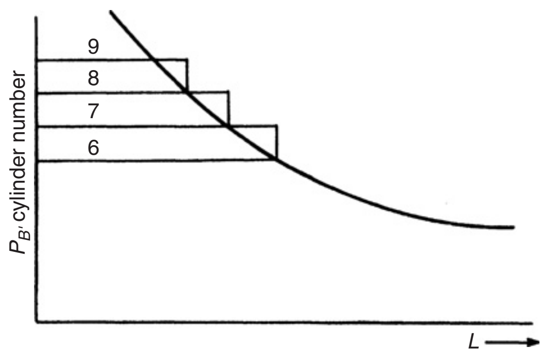


Figure 9.52 Propulsion power P_B and corresponding engine cylinder number as a function of ship's length.

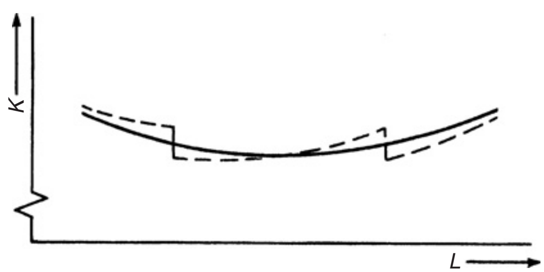


Figure 9.53 Effect of a change in number of engine cylinders on the cost of the ship.

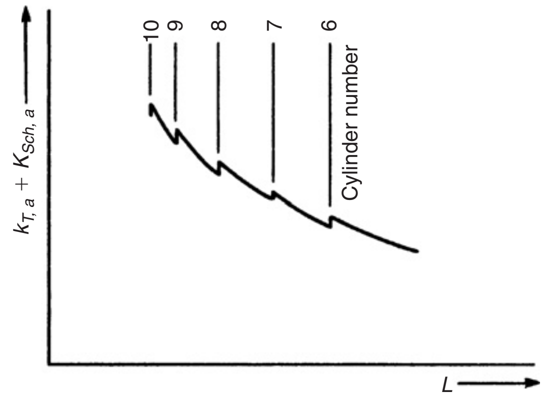


Figure 9.54 Annual fuel and lubricant costs ($k_f + k_l$) as a function of number of engine cylinders and ship's length.

9.6.5 Discussion of some important parameters

9.6.5.1 *Width*

A lower limit for B comes from requiring a minimum metacentric height GM and, indirectly, a maximum possible draught. The GM requirement is formulated in an inequality requiring a minimum value, but allowing larger values which are frequently obtained for tankers and bulkers.

9.6.5.2 *Length*

Suppose the length of a ship is varied while cargo weight, deadweight and hold size, but also $A_M \cdot L$, B/T, B/D and C_B are kept constant (Figure 9.55). (Constant displacement and underdeck volume, approximate constant cargo weight and hold capacity.) Then a 10% increase in length will reduce A_M by 10%. D, B and T are each reduced by around 5%. L/B and L/D are each increased by around 16%.

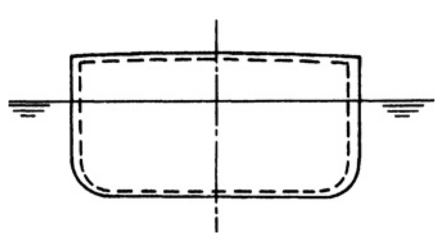


Figure 9.55 Variation of midship section area A_M with proportions unchanged.

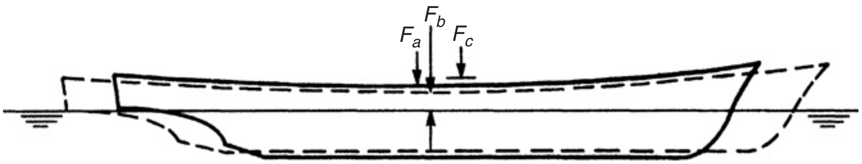


Figure 9.56 Effect of length variation on the freeboard. F_a = freeboard of basis form, F_b = freeboard of distorted ship, F_c = desired freeboard after lengthening.

For this kind of variation, increasing length has these consequences:

1. Increase in required regulation freeboard with decrease in existing freeboard.
2. Decrease in initial stability.
3. Better course-keeping ability and poorer course-changing ability.
4. Increase in steel weight.
5. Decrease in engine output and weight—irrespective of the range of Froude number.
6. Decrease in fuel consumption over the same operational distance.

Increase in the regulation freeboard
The existing freeboard is decreased, while the required freeboard is increased (Figure 9.56). These opposing tendencies can easily lead to conflicts. The freeboard regulations never conflict with a shortening of the ship, if C_B is kept constant.

Reduction in initial stability
The optimization often requires constant initial stability to meet the prescribed requirements and maintain comparability. A decrease in GM is then, if necessary, compensated by a slight increase of B/T, reducing T and D somewhat. This increases steel weight and decreases power savings.

Course-keeping and course-changing abilities
These characteristics are in inverse ratio to each other. A large rudder area improves both.

Increase in steel weight, decrease in engine output and weight, decrease in fuel consumption
These changes strongly affect the economics of the ship, see Section 9.6.4.

9.6.5.3 *Block coefficient*

Changes in characteristics resulting from reducing C_B:

1. Decrease in regulation freeboard for $C_B < 0.68$ (referred to 85% D).
2. Decrease in area below the righting arm curve if the same initial stability is used.
3. Slight increase in hull steel weight.
4. Decrease in required propulsion power, weight of the engine plant and fuel consumption.
5. Better seakeeping, less added resistance in seaway, less slamming.
6. Less conducive to port operation as parallel middlebody is shorter and flare of ship ends greater.
7. Larger hatches, if the hatch width increases with ship width. Hatch covers therefore are heavier and more expensive. The upper deck area increases.
8. Less favourable hold geometry profiles. Greater flare of sides, fewer rectangular floor spaces.
9. The dimensional limits imposed by slipways, docks and locks are reached earlier.
10. Long derrick and crane booms, if the length of these is determined by the ship's width and not the hatch length.

Initial stability
GM remains approximately constant if B/T is kept constant. However, the prescribed GM is most effectively maintained by varying the width using Mühlbradt's formula:

$$B = \frac{B_0}{C[(C_B/C_{B0})^2 - 1] + 1}$$

$C = 0.12$ for passenger and containerships
$C = 0.16$ for dry cargo vessels and tankers.

Seakeeping
A small C_B usually improves seakeeping. Since the power requirement is calculated for trial conditions, no correction for the influence of seastate is included. Accordingly, the optimum C_B for service speed should be somewhat smaller than that for trial speed. There is no sufficiently simple and accurate way to determine the power requirement in a seastate as a function of the main dimensions. Constraints or the inclusion of some kind of consideration of the

seakeeping are in the interest of the ship owner. If not specified, the shipyard designer will base his optimization on trial conditions.

Size of hold

For general cargo ships, the required hold size is roughly constant in proportion to underdeck volume. For container and Ro-Ro ships, reducing C_B increases the 'noxious spaces' and more hold volume is required.

Usually the underdeck volume $\nabla_D = L \cdot B \cdot D \cdot C_{BD}$ is kept constant. Any differences due to camber and sheer are either disregarded or taken as constant over the range of variation. C_{BD} can be determined with reasonable accuracy by empirical equations:

$$C_{BD} = C_B + c \cdot \left(\frac{D}{T} - 1\right) \cdot (1 - C_B)$$

with $c = 0.3$ for U-shaped sections and $c = 0.4$ for V-shaped sections. See also Section 9.2.4.2.

With the initial assumption of constant underdeck volume, the change in the required engine room size, and any consequent variations in the unusable spaces at the ship's ends and the volume of the double bottom are all initially disregarded. A change in engine room size can result from changes in propulsion power and in the structure of the inner bottom accommodating the engine seatings.

The effect on cost

A C_B variation changes the hull steel and propulsion system costs. Not only the steel weight, but also the price of the processed tonne of steel is variation-dependent. A tonne of processed steel of a ship with full C_B is relatively cheaper than that of a vessel with fine C_B. See also Carreyette formula, Carreyette (1978).

The specific costs of hull steel differ widely over the extent of the hull. We distinguish roughly the following categories of difficulty:

1. Flat areas with straight sections in the parallel middlebody.
2. Flat areas with straight sections not situated in the parallel middlebody, e.g. a piece of deck without sheer or camber at the ship's ends. More work results from providing an outline contour adapted to the outer shell and because the shortening causes the sections to change cross-section also.
3. Slightly curved areas with straight or curved sections. The plates are shaped locally using forming devices, not pre-bent. The curved sections are pre-formed.
4. Areas with a more pronounced curvature curved only in one direction, e.g. bilge strake in middlebody. The plates are rolled cold.
5. Medium-curved plates curved multidimensionally, e.g. some of those in the vicinity of the propeller aperture. These plates are pressed and rolled in various directions when cold.
6. Highly curved plates curved multidimensionally, e.g. the forward pieces of bulbous bows. These plates are pressed or formed when hot, or heat line bending used.

Decreasing C_B complicates design and construction, thus increasing costs:

1. More curved plates and sections, fewer flat plates with rectangular boundaries.
2. Greater expenditure on construction details.
3. Greater expenditure on wooden templates, fairing aids, gauges, etc.
4. More scrap.
5. More variety in plates and section with associated costs for storekeeping and management.

An increase in C_B by $\Delta C_B = 0.1$ will usually increase the share of the weight attributable to the flat areas of the hull (group (1) of the above groups) by 3%. About 3% of the overall hull steel will move from groups (3)–(5) to groups (1) and (2). The number of highly curved plates formed multidimensionally (group (6)) is hardly affected by a change in C_B. The change in weight of all curved plates and sections of the hull depends on many factors. It is approximately $0.33\Delta C_B \cdot$ hull steel weight.

9.6.5.4 *Speed*

The speed can be decisive for the economic efficiency of a ship and influences the main dimensions in turn. Since speed specifications are normally part of the shipping company requirements, the shipyard need not give the subject much consideration. Since only the agreement on trial speed, related to smooth water and full draught, provides both shipyard and shipping company with a clear contractual basis, the trial speed will be the normal basis for optimization. However, the service speed could be included in the optimization as an additional condition. If the service speed is to be attained on reduced propulsion power, the trial speed on reduced power will normally also be stated in the contract. Ships with two clearly defined load conditions can have both conditions considered separately, i.e. fully loaded and ballast.

Economic efficiency calculations for the purpose of optimizing speed are difficult to formulate due to many complex boundary conditions. Schedules in a transport chain or food preservation times introduce constraints for speed. (For both fish and bananas, for example, a preservation period of around 17 days is assumed.)

Speed variation may proceed on two possible assumptions:

1. Each ship in the variation series has *constant transportation capacity*, i.e. the faster variant has smaller carrying capacity.
2. Each ship in the variation series has a *constant carrying capacity*, i.e. the faster variant has a greater transportation capacity than the slower one and fewer ships are needed.

Since speed increase with constant carrying capacity increases the transportation capacity, and a constant transportation capacity leads to a change of ship size, it is better to compare the transport costs of 1 tonne of cargo for various ships on one route than to compare costs of several ships directly.

Essentially there are two situations from which an optimization calculation can proceed:

1. Uncompetitive situation. Here, speed does not affect income, e.g. when producer, shipping company and selling organizations are under the same ownership as in some areas of the banana and oil business.
2. Competitive situation. Higher speed may attract more cargo or justify higher freight rates. This is the prime reason for shipowners wanting faster ships. Both available cargo quantity and freight rate as a functions of speed are difficult to estimate.

In any case, all variants should be burdened with the interest on the tied-up capital of the cargo. For the uncompetitive situation where the shipowner transports his own goods, this case represents the real situation. In the competitive case, it should be a lower limit for attractiveness of the service. If the interest on cargo costs are not included, optimizations for dry cargo vessels usually produce speeds some 2 knots or more below normal.

Closely related with the question of optimum speed is that of port turnaround times. Shortening these by technical or organizational changes can improve the ship's profitability to a greater extent than by optimizing the speed.

Some general factors which encourage higher ship speeds are, Buxton (1972, 1976)

- High-value cargo.
- High freight rates.
- Competition, especially when freight rates are fixed as in Conferences.
- Short turn-around time.
- High interest rates.
- High daily operating costs, e.g. crew.
- Reduced cost of machinery.
- Improved hull form design, reduced power requirements.
- Smoother hulls, both new and in service, e.g. by better coatings.
- Cheap fuel.
- Lower specific fuel consumption.

9.6.6 Special cases of optimization

9.6.6.1 *Optimization of repeat ships*

Conditions for series shipbuilding are different from those for single-ship designs. Some of the advantages of series shipbuilding can also be used in repeat ships. For a ship to be built varying only slightly in size and output from a basis ship, the question arises: 'Should an existing design be modified or a new design developed?' The size can be changed by varying the parallel middlebody. The speed can be changed by changing the propulsion unit. The economic efficiency (e.g. yield) or the initial costs have to be examined for an optimum new design and for modification of an existing design.

The advantages of a repeat design (and even of modified designs where the length of the parallel middlebody is changed) are:

1. Reduced design and detailed construction work can save considerable time, a potentially crucial bargaining point when delivery schedules are tight.
2. Reduced need for jigs for processing complicated components constructed from plates and sections.
3. Greater reliability in estimating speed, deadweight and hold size from a basis ship, allowing smaller margins.
4. Greater accuracy in calculating the initial costs using a 'cost difference' method.

Where no smaller basis ship exists to fit the size of the new design, the objective can still be reached by shortening a larger basis ship. This reduces C_B. It may be necessary to re-define the midship area if more than the length of the parallel middlebody is removed. Deriving a new design from a basis ship of the same speed by varying the parallel middlebody is often preferable to developing a new design. In contrast, transforming a basis ship into a faster ship merely by increasing the propulsion power is economical only within very narrow limits.

Simplified construction of steel hull

Efforts to reduce production costs by simplifying the construction process have given birth to several types of development. The normal procedure employed in cargo shipbuilding is to keep C_B far higher than optimum for resistance. This increases the portion of the most easily manufactured parallel middlebody.

Blohm and Voss adopted a different method of simplifying ship forms. In 1967 they developed and built the *Pioneer* form which, apart from bow and stern bulbs, consisted entirely of flat surfaces. Despite 3–10% lower building costs, increased power requirement and problems with fatigue strength in the structural elements at the knuckles proved this approach to be a dead end.

Another simple construction method commonly used in inland vessels is to build them primarily or entirely with straight frames. With the exception of the parallel middlebody, the outer shell is usually curved only in one direction. This also increases the power requirement considerably.

Ships with low C_B can be simplified in construction – with only little increase in power requirement – by transforming the normally slightly curved surfaces of the outer shell into a series of curved and flat surfaces. The curved surfaces should be made as developable as possible. The flat surfaces can be welded fairly cheaply on panel lines. Also, there is less bending work involved. The difference between this and the *Pioneer* form is that the knuckles are avoided. C_B is lower than in the *Pioneer* class and conventional ships. Optimization calculations for simple forms are more difficult than for normal forms since often little is known about the hydrodynamic characteristics and building costs of simplified ship forms.

There are no special methods to determine the resistance of simplified ships, but CFD methods may bring considerable progress within the next decade. Far more serious is the lack of methods to predict the building costs by consideration of details of construction, Kaeding (1997).

9.6.6.2 *Optimizing the dimensions of containerships*

The width

The effective hold width of containerships corresponds to the hatch width. The area on either side of the hatch which cannot be used for cargo is often used as a wing tank. Naturally, the container stowage coefficient of the hold, i.e. the ratio of the total underdeck container volume to the hold volume, is kept as high as possible. The ratio of container volume to gross hold volume (including wing tanks) is usually 0.50–0.70. These coefficients do not take into account any partial increase in height of the double bottom. The larger ratio value applies to full ships with small side strip width and the smaller to fine vessels and greater side strip widths.

For constant C_B, a high container stowage coefficient can best be attained by keeping the side strip of deck abreast of the hatches as narrow as possible. Typical values for the width of this side strip on containerships are:

For small ships:	≈ 0.8–1.0 m
For medium-sized ships:	≈ 1.0–1.5 m
For larger ships:	≈ 1.2–2.0 m

The calculated width of the deck strip adjacent to the hatches decreases relative to the ship's width with increasing ship size. The variation in the figure also decreases with size.

If the ship's width were to be varied only in steps as a multiple of the container width, the statistics of the containership's width would indicate a stepped or discontinuous relationship. However, the widths are statistically distributed fairly evenly. The widths can be different for a certain container number stowed across the ship width, and ships of roughly the same width may even have a different container number stowed across the ship. The reason is that besides container stowage other design considerations (e.g. stability, carrying capacity, favourable proportions) influence the width of containerships. The difference between the continuous variation of width B and that indicated by the number and size of containers is indicated by the statistically determined variation in the wing tank width, typically around half a container width. The practical compromise between strength and construction considerations on the one hand and the requirement for good utilization on the other hand is apparently within this variation.

The length

The length of containerships depends on the hold lengths. The hold length is a 'stepped' function. However, the length of a containership depends not only on the hold lengths. The length of the fore peak may be varied to achieve the desired ship length. Whether the fore end of the hold is made longer or shorter is of little consequence to the container capacity, since the fore end of the hatch has, usually, smaller width than midships, and the hold width decreases rapidly downwards.

The depth

Similarly the depth of the ship is not closely correlated to the container height, since differences can be made up by the hatchway coaming height. The double bottom height is minimized because wing tanks, often installed to improve torsional rigidity, ensure enough tank space for all purposes.

Optimization of the main dimensions

The procedure is the same as for other ships. Container stowage (and thus hold space not occupied by containers) are included at a late stage of refining the optimization model. This subsequent variation is subject to, for example, stability constraints.

The basis variant is usually selected such that the stowage coefficient is optimized, i.e. the deck strips alongside the hatches are kept as narrow as possible.

If the main dimensions of the ship are now varied, given constant underdeck capacity and hold size, the number of containers to be stowed below deck will no longer be constant. So the main dimensions must be corrected. This correction is usually only marginal.

Since in slender ships the maximum hold width can only be fully utilized for a short portion of the length, a reduction in the number of containers to be stowed across the width of the midship section would only slightly decrease the number of containers. So the ratio of container volume to hold volume will change less when the main dimensions are varied on slender containerships than on fuller ships.

9.6.7 Developments of the 1980s and 1990s

9.6.7.1 *Concept exploration models*

Concept exploration models (CEMs) have been proposed as an alternative to 'automatic' optimization. The basic principle of CEMs is that of a direct search optimization: a large set of candidate solutions is generated by varying design variables. Each of these solutions is evaluated and the most promising solution is selected. However, usually all solutions are stored and graphically displayed so that the designer gets a feeling for how certain variables influence the performance of the design. It thus may offer more insight to the design process. However, this approach can quickly become impractical due to efficiency problems. Erikstad (1996) gives the following illustrating example: given ten independent design variables, each to be evaluated at ten different values, the total number of combinations becomes 10^{10}. If we assume that each design evaluation takes 1 millisecond, the total computer time needed will be 10^7 seconds – more than 3 months.

CEM applications have resorted to various techniques to cope with this efficiency problem:

- Early rejection of solutions not complying with basic requirements, Georgescu *et al.* (1990).
- Multiple steps methods where batches of design variables are investigated serially, Nethercote *et al.* (1981).
- Reducing the number of design variables, Erikstad, (1994).
- Increasing the step length.

Erikstad (1994) offers the most promising approach, which is also attractive for steepness search optimization. He presents a method to identify the most important variables in a given design problem. From this, the most influential set of variables for a particular problem can be chosen for further exploration in a CEM. The benefit of such a reduction in problem dimension, while keeping the focus on the important part of the problem, naturally increases rapidly with the dimension of the initial problem. Experience of the designer may serve as a short cut, i.e. select the proper variables without a systematic analysis, as proposed by Erikstad.

Among the applications of CEM for ship design are:

- A CEM for small warship design, Eames and Drummond (1977), based on six independent variables. Of the 82 944 investigated combinations, 278 were acceptable and the best 18 were fully analysed.
- A CEM for naval SWATH design, Nethercote *et al.* (1981), based on seven independent variables.
- A CEM for cargoship design, Georgescu *et al.* (1990), Wijnholst (1995), based on six independent variables.

CEM incorporating knowledge-based techniques have been proposed by Hees (1992) and Erikstad (1996), who also discuss CEM in more detail.

9.6.7.2 *Optimization shells*

Design problems differ from most other problems in that from case to case different quantities are specified or unknown, and the applicable relations may change. This concerns both economic and technical parts of the optimization model. In designing scantlings for example, web height and flange width may be variables to be determined or they may be given if the scantling continues other structural members. There may be upper bounds due to spatial limitations, or lower bounds because crossing stiffeners, air ducts, etc. require a structural member to be a certain height. Cut-outs, varying plate thickness, and other structural details create a multitude of alternatives which have to be handled. Naturally most design problems for whole ships are far more complex than the sketched 'simple' design problem for scantlings.

Design optimization problems require in most cases tailor-made models, but the effort of modifying existing programs is too tedious and complex for designers. This is one of the reasons why optimization in ship design has been largely restricted to academic applications. Here, methods of 'machine intelligence' may help to create a suitable algorithm for each individual design problem. The designer's task is then basically reduced to supplying:

- a list of specified quantities;
- a list of unknowns including upper and lower bounds and desired accuracy;
- the applicable relations (equations and inequalities).

In conventional programming, it is necessary to arrange relations such that the right-hand sides contain only known quantities and the left-hand side only one unknown quantity. This is not necessary in modern optimization shells. The relations may be given in arbitrary order and may be written in the most convenient way, e.g. $\nabla = C_B \cdot L \cdot B \cdot T$, irrespective of which of the variables are unknown and which are given. This 'knowledge base' is flexible in handling diverse problems, yet easy to use.

Such optimization shells include CHWARISMI, Söding (1977), and DELPHI, Gudenschwager (1988). These shells work in two steps. In the first step the designer compiles all relevant 'knowledge' in the form of relations. The shell checks if the problem can be solved at all with the given relations and which of the relations are actually needed. Furthermore, the shell checks if the system of relations may be decomposed into several smaller systems which can be solved independently. After this process, the modified problem is converted into a Fortran program, compiled and linked. The second step is then the actual numerical computation using the Fortran program.

The following example illustrates the concept of such an optimization shell. The problem concerns the optimization of a containership and is formulated for the shell in a quasi-Fortran language:

```
      PROGRAM CONT2
C Declaration of variables to be read from file
C TDW      t        deadweight
C VORR     t        provisions
C VDIEN    m/s      service speed
C TEU      -        required TEU capacity
C TUDMIN            share of
                    container capacity underdeck (<1.)
C NHUD              number of bays under deck
C NHOD              number of bays on deck
C NNUD              number of stacks under deck
C NNOD              number of stacks on deck
C NUEUD             number of tiers under deck
C MDHAUS   t        mass of deckhouse
C ETAD     -        propulsive efficiency
C BMST     t/m**3   weight coefficient for hull
C BMAUE    t/m**2   weight coefficient for E&O
C BMMA     t/kW     weight coefficient for engine
C BCST     DM/t     cost per ton steel hull
C BCAUE    DM/t     cost per ton E&O (initial)
C BCMA     DM/t     cost per ton engine (initial)
C
C Declaration of other variables
C LPP      m        length between perpendiculars
C BREIT    m        width
C TIEF     m        draft
C CB                block coefficient
C VOL      m**3     displacement volume
C CBD               block coefficient related to
                    main deck
C DEPTH    m        depth
C LR       m**3     hold volume
C TEUU              number of containers under deck
C TEUO              number of containers on deck
C NUEOD             number of tiers on deck
C GM       m        metacentric height
C PD       kW       delivered power
C MSTAHL   t        weight of steel hull
C MAUE     t        weight of E&O
C MMASCH   t        machinery weight
C CSCHIF   DM       initial cost of ship
C CZUTEU   DM/TEU   initial cost/carrying capacity
C
C Declare type of variables
   REAL BCAUE, BCMA, BCST, BMAUE, BMMA, BMST, ETAD,
   MDHAUS, REAL TEU, TDW, TUDMIN, VDIEN, VORR REAL
   NHOD, NHUD, NNOD, NNUD, NUEUD
C Input from file of required values
   CALL INPUT (BCAUE, BCMA, BCST, BMAUE, BMMA, BMST,
   ETAD, MDHAUS, & TDW, TEU, TUDMIN, VDIEN, VORR, NHOD,
   NHUD, NNOD, NNUD, NUEUD)
C unknowns     start    initial   lower     upper
C              value    stepsize  limit     limit
UNKNOWNS  LPP   (120.,     20.0,      50.0,    150.0),
 &        BREIT (20.,       4.0,      10.0,     32.2),
 &        TIEF  (5.,        2.0,       4.0,      6.4),
 &        CB    (0.6,       0.1,       0.4,     0.85),
 &        VOL   (7200.,   500.0,    1000.0,  30000.0),
 &        CBD   (0.66,      0.1,        .5,     0.90),
 &        DEPTH (11.,       2.0,       5.0,     28.0),
 &        LR    (12000.,  500.0,   10000.0,  50000.0),
 &        TEUU  (.5*TEU,   20.0,       0.0,     TEU ),
 &        TEUO  (.5*TEU,   20.0,       0.0,     TEU ),
 &        NUEOD (2.,         .1,       1.0,      4.0),
 &        GM    (1.0,       0.1,       0.4,      2.0),
 &        PD    (3000.,   100.0,     200.0,  10000.0),
 &        MSTAHL(1440.    100.0,     200.0,  10000.0),
 &        MAUE  (360.,     50.0,      50.0,   2000.0),
 &        MMASCH(360.,     50.0,      50.0,   2000.0).
 &        CSCHIF(60.E6,    1.E6,      2.E6,    80.E6),
 &        CZUTEU(30000.   5000.,    10000.,  150000.)
C **** Relations decribing the problem ****
C mass and displacement
   VOL         =  LPP*BREIT*TIEF*CB
   VOL*1.03    =  MSTAHL + MAUE + MMASCH 1 TDW
   MSTAHL      =  STARUM (BMST,LPP,BREIT,TIEF,DEPTH,CBD)
   MAUE        =  BMAUE*LPP*BREIT
   MMASCH      =  BMMA*(PD/0.85)**0.89
C stability
   GM          =  0.43*BREIT - (MSTAHL*0.6*DEPTH
  &                            +MDHAUS*(DEPTH+6.0)
  &                            +MAUE*1.05*DEPTH
  &                            +MMASCH*0.5*DEPTH
  &                            +VORR*0.4*DEPTH
  &                            +TEUU*MCONT*
                               (0.743-0.188*CB)
  &                            +TEUO*MCONT*(DEPTH +
                               2.1+0.5*NUEOD*HCONT))
  &                            /VOL/1.03
C  hold
   CBD         =  CB+0.3*(DEPTH-TIEF)/TIEF*(1.2 CB)
   LR          =  LPP*BREIT*DEPTH*CBD*0.75
C container stowing/main dimensions
   LPP   .GE.  (0.03786+0.0016/CB**5)*LPP
  &            +0.747*PD**0.385
  &            +NHUD*(LCONT+1.0)
  &            +0.07*LPP
   LPP   .GE.  0.126*LPP+13.8
  &            +(NHOD-2.)*(LCONT+1.0)
  &            +0.07*LPP
   BREIT .GE. 2.*2.0+BCONT*NNUD+(NNUD+1.)*0.25
   BREIT .GE. 0.4+BCONT*NNOD+(NNUD-1)*0.04
   DEPTH .GE. (350+45*BREIT)/1000.+NUEUD*HCONT
              -1.5
   TEU  = TEUU+TEUO
   TEUU .GE. TUDMIN*TEU
   TEUU = (0.9*CB+0.26) *NHUD*NNUD*NUEUD
   TEUO = (0.5*CB+.55) *NHOD*NNOD*NUEOD
C  propulsion
   PD          = VOL**0.567*VDIEN**3.6/(153.*ETAD)
C building cost
   CSCHIF      = BCST*MSTAHL*SQRT(.7/CB)
                 +BCAUE*MAUE + BCMA*MMASCH
   CZUTEU      = CSCHIF/(TEUU+TEUO)
C freeboard approximation
   DEPTH - TIEF . GE. 0.025*LPP
C L/D ratio
   LPP/DEPTH.GE.8.
   LPP/DEPTH.LE.14.
C Criterion: minimize initial cost/carried container
   MINIMIZE CZUTEU
   SOLVE
```

```
C Output
   CALL OUTPUT (LPP, BREIT, TIEF, CB, VOL, CBD, DEPTH,
   LR, TEUU, TEUO,NUEOD,
   &            GM,PD,MSTAHL, MAUE,MMASCH, CSCHIF,
                CZUTEU)
   END

   REAL FUNCTION STARUM (BMST, LPP,B,T,D,CBD)
C weight of steel hull following SCHNEEKLUTH, 1985
   REAL B, BMST, CBD, C1, D, LPP, T, VOLU
   VOLU = LPP*B*D*CBD
   C1 = BMST* (1.+0.2E-5*(LPP-120.)**2)
   STARUM = VOLU*C1
 &            *(1.+0.057*(MAX(10.,LPP/D)-12.))
 &            *SQRT(30./(D-14.))
 &            *(1.+0.1*(B/D-2.1)**2)
 &            *(0.92+(1.-CBD)**2)
 END
```

The example shows that the actual formulation of the problem is relatively easy, especially since it can be based on existing Fortran procedures (steel weight in this example).

Even an optimization shell is not foolproof and errors occur frequently when beginners start using the shell. Not the least of the problems is that users formulate problems which allow no solution as improper constraints are imposed.

Another problem is that, in reality, many design problems are not so clearly defined. While there are, in principle, techniques to include uncertainty in the optimization (other than through sensitivity analyses), e.g. Schmidt (1996), extended functionality always comes at the price of added complexity for the user, which in our experience at present prevents acceptance.

Optimization shells of the future should try to extend functionality without sacrificing user-friendliness. Perhaps further incorporation of knowledge-based techniques, namely in formulating and interpreting results, could be the path to a solution. But even the most 'intelligent' system will not relieve the designer of the task to think and to decide.

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